POWER

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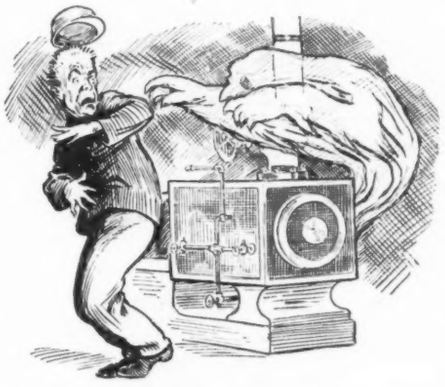
Purposeful Anecdotes



Not to Know Is to Fear

THREE gray-headed engineers were discussing the recent examination of a younger man. "I tell you," said one, "those are catch questions, and the examiner should be called down for using them."

"On the contrary, John," said another, "I feel that those questions are fair. There is nothing mysterious about the subjects, and the man who knows the why and the wherefore of these things is not going to be fooled by such questions.

"On the other hand, the man to whom these things are mysterious should not receive a license. No man is a real engineer until he has conquered the feeling that there is something supernatural about power plant equipment. Until he does he is afraid of it. The examiner was trying his man out on this basis and he was quite right."

A man becomes an engineer deserving the name, just in proportion as he masters the secrets about his work. Many a man can turn back to a moment when, because he had resolutely conquered some mystifying thing, a new power and a new outlook came to him.

When a pump balks, when an engine pounds, when a generator will not generate, **there is a reason.** The engineer believes that. It is barely possible he may not find the cause—he may have to call expert assistance, but that need not shake his faith in himself, and so long as that faith holds he has the confidence that strong men have and need.

Not long ago a certain engine stopped without being bid, and the engineer could not induce it to start. An expert was called. When he arrived he found the engineer in a nervous funk. He found a slipped eccentric, too. Ludicrous, wasn't it? Commonly in such cases the man tries to hide his ignorance behind a mask of profanity.

Cussedness was never built into a pump. Shaft governors do not race from malice. Walking under a ladder will not start a hot box. Dirty oil will. So away with mystery.

[Contributed by William E. Dixon, Norwood, Mass.]

Gas Engines Replace a Steam Turbine Power Plant

By C. L. Follmer

SYNOPSIS—Bad boiler-feed and condensing water was responsible for frequent shutdowns in the steam plant. The smaller amount of water required for the gas engine plant with the consequent smaller cost of treating and cooling was a deciding factor. Three engines are installed, each driving a 1,250-kw. generator.

In 1906 the Fairmont & Clarksburg Traction Co., now known as the Monongahela Valley Traction Co., erected on the West Fork branch of the Monongahela River, near Fairmont, W. Va., a 3000-kw. steam-turbine plant. This supplied power primarily for the local and interurban traction lines of the company. With the extension of these lines the service was also extended, taking in, particularly, the mining industry of the Fairmont region.

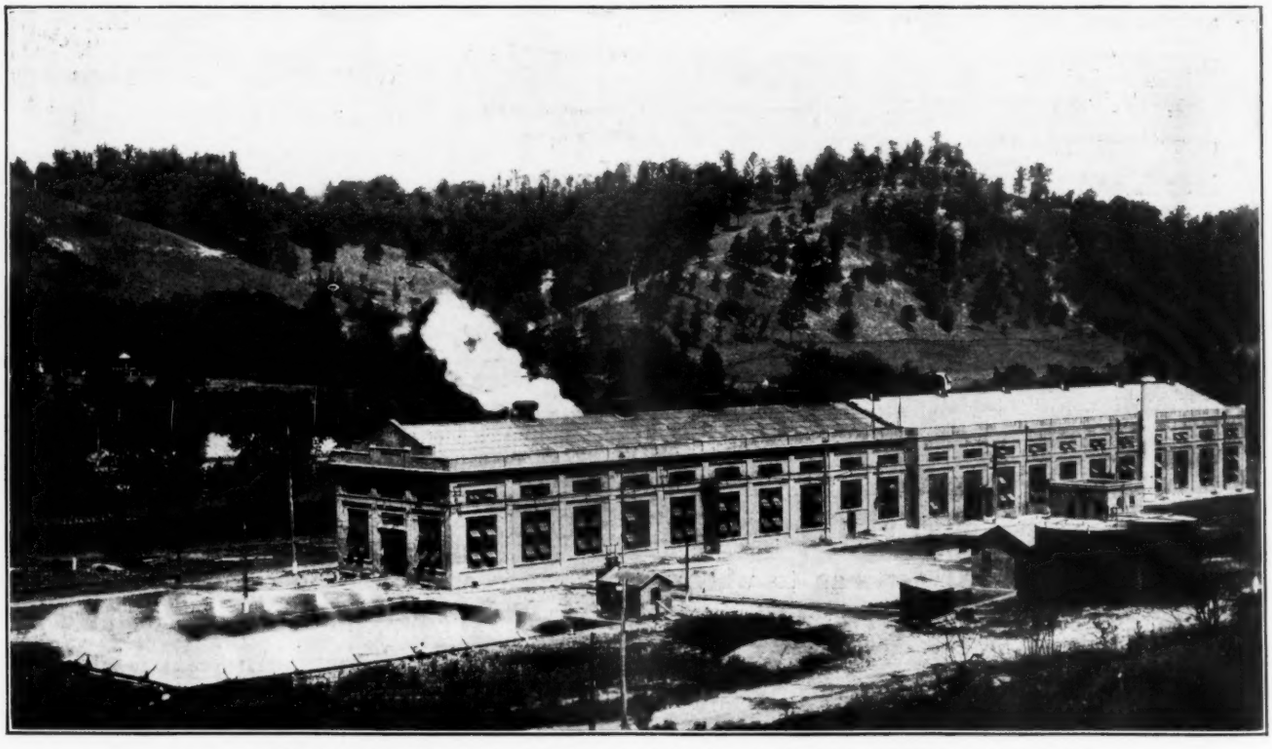


FIG. 1. COMBINED POWER HOUSE WITH COOLING POND AND PURIFICATION PLANT

After about five years' successful operation troubles in various forms began, slight at first, but gradually becoming more serious every year. The West Fork River passes forty or fifty mines and several chemical plants before reaching the power plant, and with the development of the mines and the expansion of these chemical plants, it has become highly contaminated with acids. Provision had not been made by the designing engineers to meet this problem; they had specified equipment that could not stand up under acid water and at the same time provided no means of purifying it.

Boiler compounds had always been used abundantly and served to check the attack on the boilers, but did not prevent serious shutdowns due to inability to keep enough boilers in operating condition to carry the load. A well 500 ft. deep was drilled in the hope of securing suitable boiler-feed water, but the water obtained was of a salty nature and unfit for boiler use. Various methods of water treatment and cooling were considered, as well as certain changes in the type of condensing equipment to better cope with the situation; but, for reasons outlined later, it was decided to discontinue further expenditure at this plant in attempts to relieve the situation.

The plant was not designed with any thought of extension beyond the size to which it had grown, the arrangement being such that it would be difficult and very expensive to install additional units; the power demand would prevent any radical change in the type of equipment, as the plant could not be out of commission for the length of time required to make the change; conditions were not suitable for the installation of cooling equipment, and the cost of securing condensing water from another source was deemed prohibitive.

A careful consideration of these facts convinced the officials that, if they were to maintain the company's reputation for good service, it would be necessary to provide some other source of power.

Both coal and natural gas are available in abundance, as the lines of the company pass through the heart of the coal and gas district of northern West Virginia. It was necessary, therefore, to choose between a steam-turbine plant, using either coal or natural gas under boilers, and a gas-engine plant using natural gas. The steam-turbine

plant had in its favor the following points: A much lower first cost, which, with gas-fired boilers, would be from $15 to $40 per kilowatt less than that of a gas-engine plant of the same capacity and an equal number of units; a much lower maintenance cost when general conditions are favorable; a minimum operating-labor cost in the case of gas-fired boilers; greater reliability; and proven successful operation of units in parallel.

The greatest advantage of the gas-engine plant, and one which alone might be said to balance the disadvantages, is that of a much higher thermal efficiency with a consequently much lower fuel cost. Depending on conditions, the amount of gas per kilowatt-hour in a steam plant would be from 1½ to 3 times that required in a plant using gas engines. In this particular instance the question of water-supply was most important in deciding on the type of plant. The only available supply of sufficient size in the region where it would have been desirable to locate the plant is the Monongahela River or its West Fork branch. As before stated, this water is badly contaminated with acid, and it could not be expected that the conditions would improve. Whatever type of plant was built, it would be necessary to treat the water, and it would be cheaper to cool the water for repeated use than to waste it when once treated. Here the gas-engine plant offered decided advantages. The amount of circulating water at a mean temperature of 70 deg. in high-vacuum condensing systems is about six times that required for gas-engine jackets. The loss by evaporation in cooling, however, on account of the higher temperature in the jackets and the greater amount of cooling to be effected, is from two to three times greater in the case of gas engines. For this reason the actual amount of makeup water in the condensing system is only from two to three times that in the case of gas engines, and the cost of makeup water treatment is greater in the same ratio.

Another advantage of the gas engine is that it will operate as efficiently with inlet cooling water at 100 deg. as at a lower temperature, and a difference of five or ten degrees either way does not greatly affect the amount of water to be supplied. In condensing systems high vacuum would be impossible with water at that temperature, and the highest attainable would require a very large quantity of water. The cost of pumping cooling water in either type of plant is about the same, as the increased amount pumped in the case of condensers about balances the increased head in the case of gas engines. The cost of cooling the discharge water, however, is from two to three times as much for a condensing system.

In regard to reliability, it may be said that under the same general conditions, that of gas-engine service cannot be compared with that of the steam turbine. To offset this it is usually considered expedient to install such a number of units in a gas-engine plant that one may be out of service at all times for inspection and repairs.

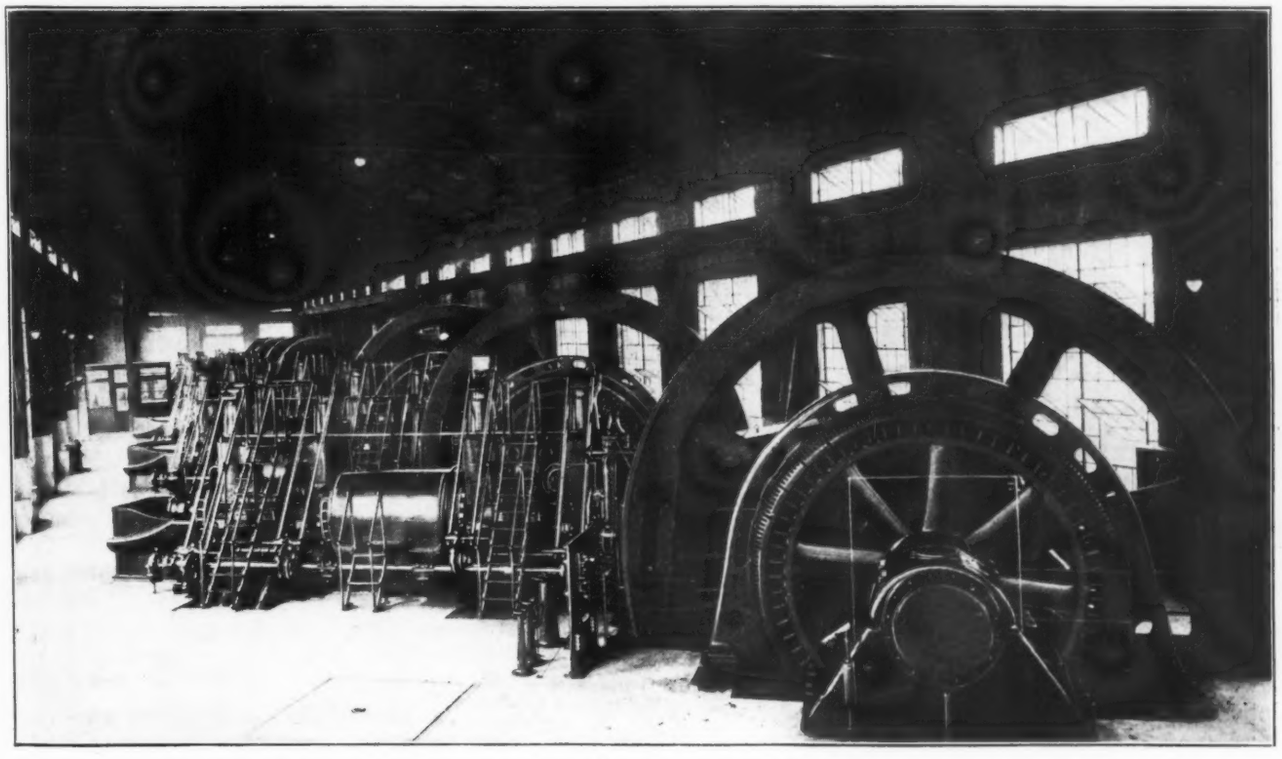


FIG. 2. NEW GAS ENGINE UNITS SET AT AN ANGLE WITH CENTER LINE OF BUILDING

A careful weighing of the advantages and disadvantages of the two types of plants resulted in the decision to install a gas-engine plant. The location in a measure was determined along with the decision on the type. The Consolidation Coal Co., Inc., has in operation at Hutchinson, W. Va., a natural-gas engine plant containing four twin-tandem Bethlehem Steel Co. engines, 23½x32-in., direct-connected to 1,150-kw. generators. Conditions were such that at the time of the troubles with the steam-turbine plant, this plant was able to furnish considerable power to the traction company. Hutchinson is centrally situated with respect to the operations of the traction company and was therefore a desirable location for the new plant. It was realized that it would be to the mutual

advantage of both companies to have the plants as near together as possible, and the new one was accordingly built adjacent to that of the coal company.

The building housing the coal company's engines is 49x190 ft., and the new building is 57x209 ft., making an overall length of practically 400 ft. The increase in width was occasioned by the installation of a slightly different type of engine from that which was installed in the other plant.

Installation of Gas Engines and Generators

The new building has sufficient room for four engines, but is provided with a temporary end with the idea of a future extension to a maximum of six engines. Foundations are in for the four, but only three engines have been installed. These are of the single-tandem type, 34x50 in., direct-connected to 1,250-kw. generators. The rated speed is 116.2 r.p.m., which is much lower than the speed of the coal company's engines of 162½ r.p.m. The weight of the flywheels is 77½ tons each, and the total weight of the engine without the flywheel is 210 tons. Each foundation contains 450 cu.yd. of concrete.

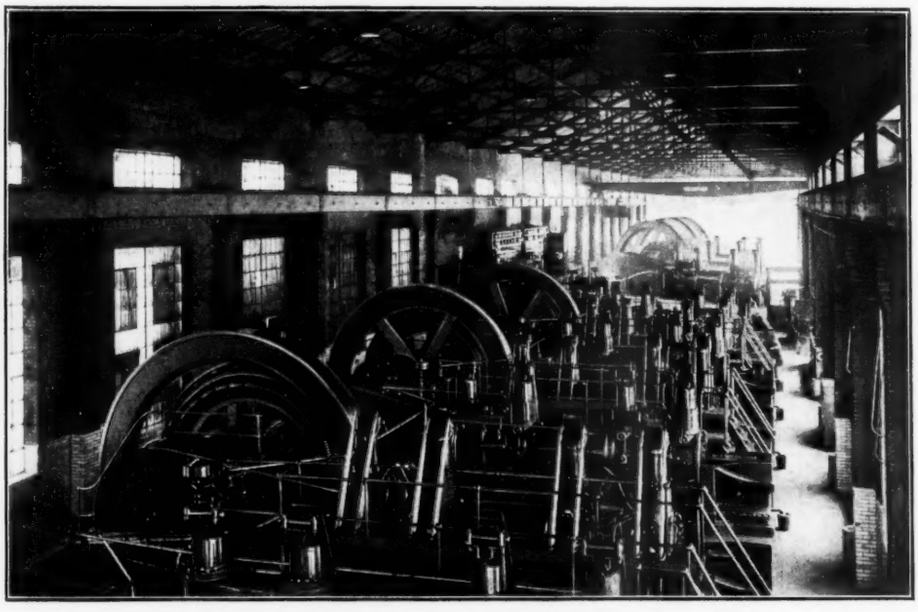


FIG. 3. ENGINE ROOM, SHOWING ORIGINAL UNITS

An interesting feature of the foundations is that, instead of the entire foundation being carried to solid rock, it rests on concrete piers. There are seven of these per foundation, at the points where the weight of the engine is carried. The main foundation extends about 3 ft. below the basement floor, the piers running from this point through a layer of quicksand and shale to the rock, 11 ft. 6 in. below. This method of support is not in general use for foundations for this type of machinery and was not recommended by the engine builders. Great care was taken in binding the foundations together in the 3-ft. space below the basement floor, and the operation of the engines thus far without vibration points to the success of the method.

Another interesting feature in the setting of the engines is that instead of the center lines being perpendicular to the center line of the building, they are at an angle of approximately 37 deg. from the perpendicular. Thus the width of the building was decreased at least 14 ft., and although the length was somewhat increased, the cost was considerably less for the same floor space.

A 20-ton crane having a span of 49 ft. was purchased for the coal company's plant, but on account of the change in type and weight of engines, a crane of 40 tons' capacity was required for the new plant, with a span of 54 ft. 10 in. This is equipped with 125-volt motors, which receive energy directly from the exciter busbars. Three of them—the ones on the bridge, the main hoist and the 5-ton auxiliary hoist—are triplicates, 25 hp. each; the trolley motor is 10 hp.

Electrical Apparatus and Auxiliaries

The switchboard is continuous with the old board, making in appearance a single unit for the two plants, blank panels being inserted at the dividing point.

The transformer room is an addition to the main building, directly back of the switchboard; it is large enough for six oil-insulated, water-cooled 2,300/23,000-volt, 1,500-kv.-a. single-phase transformers, of which three are at present installed.

Adjacent to this room but entirely apart from it is another addition, 19x19 ft., which will accommodate two 300-kw. rotary converters. These will supply direct current at 500 volts to the trolley wire that passes the plant.

The coal company's plant was provided with an exciter driven by a 115-hp. gas engine which is used for starting purposes only, and a motor-generator exciter set, used for ordinary operation. A motor-generator exciter set only was secured for the new plant. This consists of a 150-hp. 2,300-volt induction motor and 100-kw. 125-volt direct-current generator, operating at 900 r.p.m. A duplicate of this outfit will be secured later when the plant is extended.

A 13¼x6x10-in. two-stage air compressor, supplying compressed air for engine starting, is also connected by a friction clutch coupling to the 115-hp. gas engine. The compressor for the new plant is motor-driven. This is a 14x6¾x10-in. two-stage duplex machine, operating at 210 r.p.m. against 250-lb. pressure.

To save the space usually taken up by air tanks, a 16-in. extra-heavy pipe 118 ft. long, with Van Stone joints and welded necks, was installed to act as a reservoir. This pipe, together with the water-supply and drain pipes and the gas main, is between the engine foundations and the basement side walls. On the other side of the building the corresponding space is occupied by the generator cables.

The air compressor, as also the circulating water and spray pumps, is in the basement. There are two centrifugal circulating pumps, each with a capacity of 1,000 gal. per min. against a 75-ft. head and driven by a 35-hp. 2,300-volt induction motor at 1,700 r.p.m. Either of these will supply sufficient water for three engines. Pro-

vision is made for a third pump of the same size, when the plant is extended. The same idea was followed out in the spray-pump installation. There are now two De Laval centrifugal spray pumps with a capacity of 2,175 gal. per min. each against a 26-ft. head, driven by 20-hp. motors at 1,200 r.p.m., with provision for a third unit. As the basement floor is 7 ft. below the level of the water in the cooling pond, all the pumps are self-priming.

The circulating pumps receive their water through a 12-in. cast-iron pipe from the pond and discharge into a pipe of the same size, from which the engines are supplied by 5-in. leads. The main supply pipe is reduced to 10 in., 8 in. and finally 6 in. as the amount of water to be handled is decreased. The drain from the engines is also of cast iron and increases in size from 6 in. to 12 in. diameter. Another 12-in. cast-iron pipe from the cooling

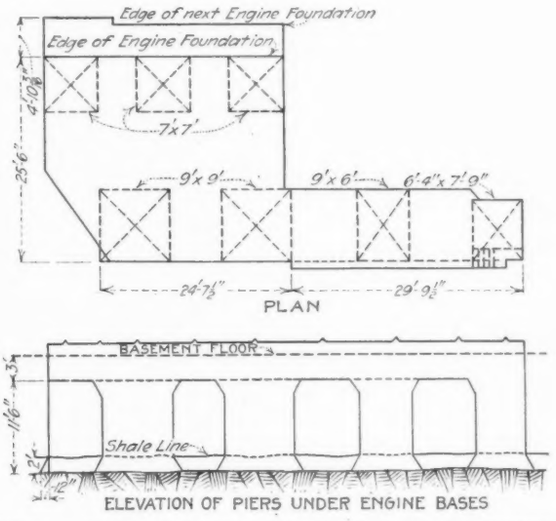


FIG. 4. PLAN AND ELEVATION OF ENGINE PIERS

pond connects with the 12-in. drain near the spray pumps, from which point the supply pipe to these pumps is increased to 16 in. diameter. This 12-in. pipe from the pond serves also as a continuation of the main drain to the pond when the sprays are not in operation and as a makeup pipe, as explained later, when the sprays are operated. The discharge to the sprays is a 16 in. pipe.

The cooling pond is 85 ft. wide by 97 ft. long by 9 ft. 6 in. deep, with sloping walls; the walls, bottom, piers for pipe support and the suction wells being of concrete. There are seventy-five ⅞-in. bronze nozzles with a capacity of 58 gal. per min. each at 7 lb. pressure, arranged in fifteen groups of five each, spaced 13 ft. centers. On the basis that the water from the engines attains a temperature of 180 deg., these sprays are guaranteed to cool approximately 2,000 gal. per min. from that temperature to 100 deg. when 2,350 gal. per min. of pond water at 100 deg. is mixed and sprayed with it. Other conditions attendant on this operation are a maximum surrounding air temperature of 100 deg. and a relative humidity of 50 per cent. The maximum total loss from the pond under these conditions is guaranteed not to exceed 2½ per cent. of the water sprayed, with an average of not over 1¼ per cent. the year round.

The makeup water is pumped from the West Fork branch of the Monongahela River, which passes the plant a short distance away. A waterproof concrete dry well about 35 ft. deep and 14 ft. square inside was constructed on the bank of the river. The bottom of the well is low enough to have the pumps self-priming at all times, and all openings in the well with the exception of those for suction pipes are above highest water level. Bronze suction wall pieces were used in duplicate, and the two suction lines outside have swing joints to allow cleaning of the strainers. One pump, a duplicate of the circulating pumps in the plant, is used, conditions happening to be such that duplication was possible.

The Water Purification System

An 8-in. cast-iron line carries the river water to the purification plant. The water from this river is of an extremely variable character, and for this reason an intermittent treating system was decided upon. The system has a capacity of 12,000 gal. per hr. The two 48,000-gal. treating tanks and the housing for the two gravity filters are shown in Fig. 1. The treated water runs by gravity to the cooling pond.

A small brick house 22 ft. square, situated about 15 ft. from the main building and discernible in the photograph, has three compartments: One contains a 56-cell storage battery, which is used in case of emergency to supply power for the ignition circuit; another is for oil and grease storage; and the third for gas metering and regulating equipment.

High-pressure gas is brought to the plant in a 6-in. main, and after being reduced to a pressure of about 8 oz., is conducted through a short 12-in. wrought iron pipe to a 36-in. main that rests on piers on the basement floor. Its length is about 130 ft., and it is constructed of No. 12 gage steel plate, lap-riveted with angle-iron riveted flanges; 16-in. leads connect it to the distributors on the engines. Such a large main would be unnecessary for natural gas, but was made this size to allow a change at some future date to operation on producer gas, this point having been considered in the design of the engines.

PRINCIPAL EQUIPMENT OF HUTCHINSON PLANT OF THE MONONGAHELA VALLEY TRACTION CO.

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
3	Gas engines	Single tandem	34x50-in.	Driving main generators	116.2 r.p.m.	Bethlehem Steel Co.
3	Generators		1,250-kw.		2,300 volts, three-phase	General Electric Co.
1	Gas engine	Vertical	115-hp.	Driving exciter and compressor		Bruce-MacBeth Engine Co.
1	Motor generator	Induction motor and d.-c. generator	100-kw.	Excitation	Motor voltage, 2,300; exciter voltage, 125	
1	Compressor	Two-stage	13½x6x10-in.	Air for engine starting	Gas-engine driven (old plant)	Ingersoll-Rand
1	Compressor	Two-stage	14x6¾x10-in.	Air for engine starting	Motor-driven (new plant) 250 lb.	Chicago Pneumatic Tool Co.
1	Motor	Induction	100-hp.	Driving compressor	2,300 volts, 720 r.p.m.	General Electric Co.
2	Pumps	Centrifugal	1,000 gal. per min.	Circulating water	Motor-driven; 75-ft. head	De Laval Steam Turbine Co.
1	Crane		40-ton	Engine room		Morgan Engineering Co.
4	Crane motors				125-volt	General Electric Co.
1	Switchboard	Blue Vermont marble	11 panels new; 9 panels old			Westinghouse Elec. & Mfg. Co.
3	Transformers	Single-phase	1,500-kv.-a.	Serving transmission lines	2,300 to 23,000 volts	Westinghouse Elec. & Mfg. Co.
2	Pumps	Centrifugal	2,175 gal. per hr.	Spray service	Motor-driven; 26-ft. head	De Laval Steam Turbine Co.
1	Spray system			Cooling jacket water	2,000 gal. per min. cooled from 180 to 100 deg.	Spray Engineering Co.
1	Purification system		12,000 gal. per hr.	Treating jacket water		W. B. Scaife & Sons Co.
1	Chimney	Reinforced concrete	50-in. diam., 45 ft. high	Carry off exhaust gases		Weber Chimney Co.

The exhaust gases are conducted through cast-iron water-cooled exhaust manifolds and a short 18-in. pipe having a U-bend with connections for water sealing when the engine is out of commission, to a 36-in. firebrick-lined, reinforced-concrete exhaust duct, placed along the side of the building and built integral with the foundation. A 50-in. diameter reinforced concrete stack 45 ft. high conveys the gases from the duct to a point above the eaves of the building. It may be stated, however, from the results experienced in this case, that the use of a concrete stack cannot be recommended for this severe service.

Air is supplied to each engine from the outside of the building through an 18-in. spiral-riveted pipe. The inlet-air pipes with their hoods may be plainly seen in the photograph.

For a plant of its size the construction was accomplished in very good time; the excavation was started on Sept. 1, 1914, and the first engine was put in operation on Feb. 8, 1915.

On the basis of the maximum continuous output of 1,250 kw. per unit guaranteed by the builders, the cost of the installation of four engines complete with present building and all accessories, but exclusive of the cost of land, will approximate $75 per kilowatt.

Cold-Air Refrigeration Plant

The main building of a certain model packing house on the Pacific Coast is a five-story, reinforced-concrete structure, approximately 300x300 ft., where about 125 head of cattle, 200 head of hogs and 800 head of sheep are handled daily.

There are two beef chillrooms about 40x80 ft., two hog chillrooms about 20x80 ft., and two similar rooms for sheep. All are kept at the proper temperature by the circulation of cold air through air ducts. This air is cooled by passing it around 5,000 ft. of 2-in. ammonia pipe on the top floor or bunker room. It is then forced down ducts at one end of the building by a 5-hp. motor-driven fan, and the air returns to the ammonia coils up ducts at the other ends of the rooms, thus keeping a good circulation of fresh air in the coolers at all times.

With this system of cooling the rooms are clean and dry, there being no sweating of pipes and dripping of water. In addition to this there are a Sharp freezer and a storage freezer. In the former a temperature of from 5 to 10 deg. below zero is maintained. In the storage freezer the temperature ranges from 12 to 14 deg. above zero. These two rooms are refrigerated by direct expansion. The offal room is refrigerated by the curtain system, cold brine circulating over a system of cloth curtains. All power is furnished by induction motors.

The power plant is a separate building about 100 ft. from the main structure. The machinery consists of a 125-ton cross-compound noncondensing Corliss compressor. The high-pressure steam cylinder is 16x26 in.; low-pressure, 32x26 in.; the ammonia cylinders 13x26 in.; one compressor, with steam end, 12x30 in.; ammonia cylinder, 9x18 in.; one air compressor with steam end, 12x14 in., the air from which is used principally to pump water from a 12-in., 125-ft. deep well, the water level being about 40 ft. from the surface. This well furnishes about 100,000 gal. of water per day.

Until recently electricity for lighting was furnished by a motor-generator set, but on indicating the smaller engine and finding that it had ample power to run the generator also, the motor was discarded and the generator belted to the engine. It has been running that way for months, the saving on the cost of lights being $125 per month.

These engines have been run continually day and night for the past eight years, having been stopped only for minor repairs. They are equipped with mechanical-feed lubricators and gravity sight-feed on the bearings.

The pride of the plant is the boiler room, which is 40x60 ft. and has a 40-ft. ceiling. The equipment consists of two 145-hp. water-tube boilers set in a battery and one 300-hp. water-tube boiler. A steam pressure of 150 lb. is carried.

The boilers are fed by a 7&4½x10-in. simplex pump, pumping water from an open heater at a temperature of from 204 to 206 deg. There is also an injector on the city-water supply for emergency. Oil is used as fuel, the 300-hp. boiler being equipped with two patent furnaces, the burners being placed at the bridge-wall and throwing the fire toward the doors. Each of the 150-hp. boilers has one burner and furnace. The steam and oil pressure to the burners is controlled automatically by regulators, so that the services of a fireman are not required.

TEST OF A 300-H.P. "CLASS A" STIRLING BOILER

	Apr. 2, 1915	Apr. 3, 1915
Date of tests		
Duration of test, hr	7	8
Steam pressure, lb	147	148
Temperature of feed water, deg	204	204
Factor of evaporation	1.055	1.055
Temperature of boiler room, deg	80	82
Temperature of flue gases, deg	576	546
Draft in rear pass, in water	0.085	0.065
Carbon dioxide, per cent	12.1	12.9
Oxygen, per cent	3.9	3.3
Total water evaporated, actual, lb	76,319	87,723
Total water evaporated from and at 212 deg., lb	80,516	92,548
Water per hour evaporated from and at 212 deg., lb	11,502	11,568
Oil pressure to burners, lb	10.5	11.3
Total oil fired, lb	5,357	6,107
Oil per hour as fired, lb	765	763
Boiler horsepower developed	322	348
Per cent. above rating	8.0	16.0
Water evaporated per pound of oil, lb	14.25	14.36
Water evaporated per pound of oil from and at 212 deg., lb	15.03	15.15
Boiler efficiency, per cent	80.1	81.3

A low-water alarm not only blows a whistle in case of low water, but also shuts off the oil supply to the burners in case the alarm is not heeded by the engineer. The dampers are controlled by automatic damper regulators operated by the oil pressure from the burners. These hold the heat in the boiler and maintain a high boiler efficiency at all loads. The amount of draft is shown by differential draft gages.

Recently, two tests were run on the 300-hp. boiler—one of 7-hr. and one of 8-hr. duration, both oil and water being carefully weighed. These tests were run under regular plant conditions and the results speak for themselves. The boiler has been in service eight years and has had no repairs excepting a few tubes.

A daily engine-room log is kept. The operating crew consists of a day engineer, a night engineer, a boiler washer and a general utility man who is licensed and relieves the operating engineer one day a month.

Heating is Manifested Soonest, with the same expenditure of energy, in small bodies because they are not capable of absorbing the heat generated, therefore feel hot to the touch, while the same energy expended on large masses will produce only a slight rise in their temperature.

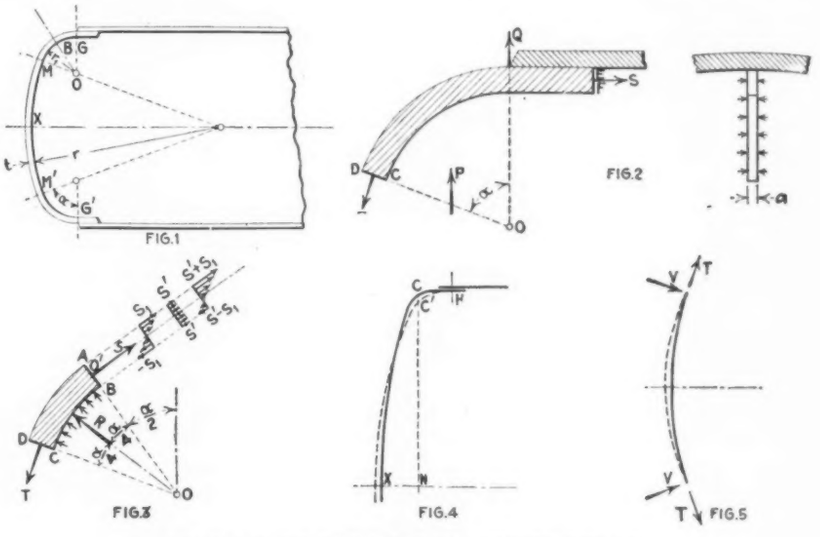
Specific Heat of Iron is 0.1138 B.t.u. per pound per degree F. rise in temperature, or lb. × deg. × 0.1138 = B.t.u. absorbed, and each B.t.u. equals 778 ft.-lb.; therefore each pound of iron heated one deg. F. has absorbed 0.1138 × 778 = 88.5364 ft.-lb. of energy.

Stresses in Convex Heads

By F. F. Couch*

SYNOPSIS—The author takes exception to analyses of stresses given in previous discussions of the subject, and maintains that the results of the formulas point out that the assumptions do not represent actual conditions, or that the assumed relations between internal stresses and external forces were not correctly stated, or both.

The discussions in *Power* during the past year regarding the stresses in convex boiler heads and dealing in particular with the stresses caused in the fillet, or heel, of the flange do not seem to agree. In the issue of Jan. 13, 1914, page 50, is shown a very instructive graphical representation of the strains, or unit deformations, as observed on the steam drum of a Babcock & Wilcox boiler. This would seem possible of but one interpretation—that shaded areas made by dotted lines were meant to represent unit elongations, or tensions, and shaded areas made by full lines to represent unit shortening of the material, or compression; and further, that the exact amount of elongation or shortening of the material at any point be represented by the length of a straight line perpendicular to the surface at the point and extending to the irregular curved boundary line. If we are to assume that the actual, or true, internal stresses are proportional to the exact deformations, by inspection it is apparent that the maximum tension occurs at the inside of the heel of the flange and that the only compression occurs at the outside of the heel, at the inside of the spherical portion of the convex head near the center, and the small amount of compression shown at the extreme left of the cylindrical boiler shell itself.



ANALYSIS OF STRESSES IN CONVEX HEADS

In the same issue of *Power* are two articles (pages 59 and 60) the purport of which would seem to be: First, as advocated by H. J. Vander Eb, to limit the minimum radius of the flange fillet to at least five times the thickness of the head; and second, in an article by T. T. Parker, to show that the factor of safety of 5, which now fulfills the requirements of many boiler laws, is too low—many manufacturers now designing heads by the common formula,

$$t = \frac{P \times R \times F}{2 \times TS}$$

in which

P = Allowable steam pressure;
R = Radius of curvature of head (r in Fig. 1);
TS = Ultimate tensile strength of material;
F = Factor of safety;

and using F equal to 8 or more—but that even with a nominal factor of safety of 8, convex heads have failed, the trouble always starting from the inside right at the heel of the flange and working outward until failure occurred.

On pages 10 to 13 of the July 7, 1914 issue, there appeared an article by F. G. Gasche in which were proposed formulas to give the maximum stresses occurring in the flange fillet both inside and outside; or, more exactly, to express values for the stresses at that particular part of the fillet which is just half way between the points of tangency to the cylindrical shell and to the spherical part of the head. This assumption as to the approximate location of the maximum stress is in keeping with experiment and with the point of rupture of heads that have failed. The results of the formulas, however, seem to disagree with the opinion of Mr. Vander Eb, as well as with the results of experiment. First, a compressive stress is shown to exist at the inside of the flange and a high tensile stress at the outside. This disagrees with the experiments on the Babcock & Wilcox boiler mentioned. The second point is that the formulas would seem to indicate that the flange radius tends to become smaller under the action of internal pressure, the natural and correct result of which would be to design the flange with a smaller radius in the beginning. Mr. Gasche advises a very small radius, but not a sharp corner, and has developed final formulas for a flange radius equal to twice the thickness of the head. This disagrees with Mr. Vander Eb, who recommends nothing less than five times that thickness. Now, as several boilers have failed at this particular point even with *nominal* factors of safety greater than 5, this would seem to be a very important point to clear up, so that possible future accidents from this source may be prevented.

The exact solution is a very difficult thing, and in all probability, if there were a rigid formula simple enough to recommend its general use, it would be pretty well known. The results of the formulas as given by

*Assistant Professor of mechanical engineering, Rutgers College, New Brunswick, N. J.

Mr. Gasche would seem to point to either or both of two things: That the assumptions upon which the formulas are based do not represent conditions; that the relations between internal stresses and external forces as based on those assumptions, are not stated correctly; or both. The present writer will endeavor to show in what particulars he disagrees with Mr. Gasche.

Figs. 1, 2 and 3 reproduce enough of Mr. Gasche's figures similarly numbered to bring out the comparisons. All equations referred to here by number are Mr. Gasche's.

Fig. 1 represents a cross-section through the axis, M and M' representing the points of tangency of the sphere $M\,X\,M'$ to the flange $G\,B\,M$, and it is for the point B, half way between G and M, that Mr. Gasche has written the equations. Fig. 4 represents the upper half of the same section, using single lines the more easily to show by dotted lines the tendency of the head to bulge out until, ultimately, the head would assume a spherical form of smaller radius passing through the points G and G', provided the material were sufficiently elastic, thus straightening out the flange radius entirely. It was Mr. Gasche's attempt, by using a very small radius and placing the rivets as near to this radius as riveting would permit, to bring the spherical head close to the support G in order to minimize this tendency to bulge. The trouble is, the flange radius has not been eliminated and the tendency still remains to straighten out this radius, causing bending; and here is the important point that is entirely ignored in all calculations—bending is much more harmful on curved beams than on straight ones, not obeying the same laws as to location of neutral axis and proportionality of stress, this effect increasing as the radius decreases.

There Must Be Equilibrium of the Forces

The whole theory by which the stresses in any structure are found is based upon the assumption that the internal stresses hold the external forces in equilibrium. Another principle is that the true internal stresses in any direction are determined, not by the forces in that particular direction only, but by the forces in three perpendicular directions. Thus, when it is stated that the thin strip of width a, shown at the right in Fig. 2, is acted upon by pairs of equal and opposite forces which do not affect the equilibrium of the body in a direction at right angles to themselves, that is true; but that the stresses in the material itself are not affected by such lateral forces is not true; and a strict analysis of this particular problem based upon the theory of elasticity would show that they have a considerable effect and should not be ignored. If we assume that such lateral forces may be ignored, it reduces down to a case of forces all in one plane, and it is upon this assumption that the formulas were based.

In Fig. 2 at section CD, only the normal force T is assumed to act—no bending moment, no shear—and the very simple relation that $S = T$ is deduced, the moment of Q about O (the point chosen) being zero, although this would seem possible only under the assumption of forces in only one plane, if the rivet were placed directly over O, the head of the rivet riding up on the flange curvature. Moreover, how there can be neither shear along DC, which must be the condition if Mr. Gasche's equations (4) and (10) are true, nor a bending moment at the same section, necessary for his equations (2) and (10) to be true, is difficult to reconcile with the fact that the spherical portion of the head near the center tends to decrease in radius as shown in Fig. 4, causing a bending moment at the center, the result of which would be compression on the inside and tension on the outside. This fact is borne out by the experiments on the Babcock & Wilcox boiler. In Fig. 5 is shown a curved beam having the dotted position caused by a uniform normal pressure on the concave side. This beam could assume the dotted position only by the help of lateral constraint at the end in addition to the forces T and T, such as shown by the forces V; for in the absence of V it would assume a circular shape of larger radius, not smaller. The assumption that there is no bending moment at the points M and M', Fig. 1, depends upon whether or not the radius of curvature at that point during internal pressure is equal to the original radius r. Judging from the experiments quoted, this point would be further out toward the center, where the tension on the inside and that on the outside are equal.

As before mentioned, S was proved equal to T from Fig. 2 by equation (2). This same value of S was then transferred down to Fig. 3 at an angle of $\frac{a}{2}$ to its position in Fig. 2. Now, if there is no shear on plane CD, which was the necessary condition in equations (4) and (10), the force S must be equal to the force T in Fig. 3, for the force R on the surface BC is symmetrical. Hence, there can be no shear at AB, as assumed in the four conditions of equilibrium of piece $ABCD$ in Fig. 3, stated and numbered just previous to equation (7). Therefore, the relation must exist, resolving T and S into components along the direction of force R, that

$$(T + S)\,sin\,\frac{a}{4} = R = 2\,apr_1\,sin\,\frac{a}{4},\ \text{or}\ T = apr_1$$

But from equation (5), T was given equal to $\frac{arp}{2}$, hence r_1 must equal $\frac{r}{2}$, and in the final assumption that $r_1 = 2\,t$, the necessary relation would be that $t = \frac{1}{4}\,r$, or one-fourth of the diameter of the boiler for all pressures and sizes.

Finally, with regard to the bending moment at section AB, which is supposed to decrease the flange radius r_1 instead of increasing it, the latter being the case if the head were to bulge out toward a spherical shape of smaller radius and also the case as observed on the Babcock & Wilcox boiler, it must be concluded that, neglecting the fact that $\frac{6\,M}{bt^2}$ does not give the stress caused by a bending moment M in a curved beam of rectangular cross-section b inches wide and t inches thick, equation (10) must be wrong from one or more of the causes previously pointed out.

Variation of Stresses with Variation of Radius

The writer can see no reason why the intensity of the stress at the flange fillet should not decrease as the radius of the fillet increases and gradually approach the stress in a hemispherical head as the radius of the flange fillet approaches in value the radius to which the spherical portion is formed, this latter radius then equaling half the diameter of the cylindrical shell. In other words, it would appear that limiting the radius of the

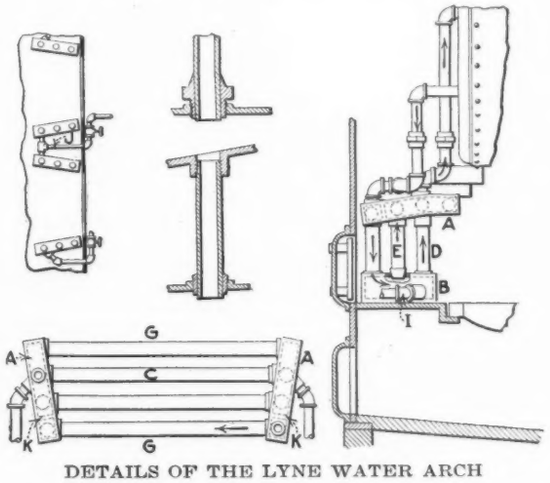
flange by a certain minimum amount is a step on the right road.

Experiments such as those conducted by Messrs. Barraclough and Gibson on the Babcock & Wilcox boiler would be very instructive if carried out upon boilers in which the flange radius was made of different lengths, everything else remaining the same, and by showing the stresses in two perpendicular directions. For, by inspection of Fig. 4, if the flange radius does tend to increase and the head to bend out, some ring cut by a plane CN, must tend to decrease in diameter as shown at C', which could result only in a compression at right angles to the plane of the paper at point C, said compression having a tendency to increase the tensile stress at the inside of the flange.

It does not seem so much an evidence that "mathematical conjecturing" is at fault, as Mr. Vander Eb expresses it, as it does that attempts are made to treat a complex case by an approximate method which does not even approach a strict mathematical analysis.

Lyne Furnace Water Arch

In this arch the water supply is received from the lower part of the front head of a horizontal tubular boiler and discharges into the same head about 2 ft. higher up. Both the supply and the discharge pipes are 1½ in. Referring to the accompanying illustration, the arch



DETAILS OF THE LYNE WATER ARCH

is made up of four cast-iron manifolds A and B connected by a series of 2-in. pipes. These manifolds are placed at angles with one another, so that the opening nearest to the fire is wider than the furnace mouth, to facilitate firing and cleaning of the fire. The roof pipes are stepped for the same reason, excepting the pipe next to the fire and the one next to it, both of which are in the same plane, so that the firebricks, which are laid on dry, may have a level support without the necessity of cutting the bricks.

There are three vertical pipes D connecting the manifolds A and B. The outside pipes are connected to the manifolds by right-hand threads at the top and left-hand threads at the bottom ends. The middle pipe E has a right-hand thread at the top, but the bottom end is made tight by a tapered copper ferrule to allow free expansion of all the tubes independently of one another and overcome unequal strains upon the tubes and castings. The hole is tapered and into it the copper ferrule is driven home and permanently calked after the outside pipes have been made up.

In the plan view the two outside roof pipes G G are attached by right-hand threads on the right ends and left-hand threads on the left ends. The inside pipes are attached in the same manner as the middle vertical pipe E. An enlarged detail of this is shown. A blowoff pipe I is provided at each side, and where two arches are united to fit two furnace mouths, as shown at J. In attaching this arch it is necessary to provide for the settling of the boiler, and to avoid straining or breaking of the pipes or fittings the equivalent of universal joints is introduced.

Water enters the upper manifold A and approximately one-half of it descends through the vertical pipe into the bottom manifold B and the other half crosses over through the front roof pipes G and down through the corresponding vertical pipe. This passage of water into the manifolds B is made compulsory by the partitions K, which divide the upper manifold into two parts. The course of water circulation is indicated by the arrows. The solid matter is swept toward the fire ends of the bottom manifolds B and is blown out through the blowoffs I before there is opportunity for it to bake on the inside of tubes and manifolds.

This water arch is the invention of Lewis F. Lyne, 39 Cortlandt St., New York City.

Gasoline from Natural Gas

By F. E. Matthews

Gasoline is now being made in many oil fields by the condensation of natural gas. The process, as described in a recent bulletin of the United States Bureau of Mines, consists of compressing the gas and then cooling it. Sometimes the cooling is carried out by water at ordinary temperatures and sometimes by the use of mechanical refrigerating systems such as are used to keep foodstuffs from spoiling. Gasoline is not a definite compound, but a more or less varying mixture of paraffin hydrocarbons.

CHEMICAL CONSTITUENTS AND COMMERCIAL PRODUCTS DERIVED FROM CRUDE PETROLEUM AND NATURAL GAS

Commercial Product, Consisting Chiefly of	Specific Gravities	Boiling Temp. C°	Boiling Temp. F°	Chemical	Name	Notes	
		—160.0	—256.0	$C\ H_4$	Methane	Occurring ordinarily in the gas.	
		— 93.0	—135.0	$C_2\ H_6$	Ethane		
		— 45.0	— 49.0	$C_3\ H_8$	Propane		
Cymogine (Butanes)	108	0.0	32.0	.....	.....		
		1.0	33.8	$C_4\ H_{10}$	Butane		
Rhigolene (Butanes and Pentanes)	92 to 94	18.0	64.4	.....	.....		
		36.0	97.0	$C_5\ H_{12}$	Pentane	Principal components of refinery gasoline.	Occurring ordinarily in the oil.
Petroleum ether (Pentanes and hexanes)	80 to 94	68.9	156.0	$C_6\ H_{14}$	Hexane		
		70 to 90.0	158 to 194	.....	.....		
Naphtha (Hexanes, heptanes, octanes)	58	98.3	209.0	$C_7\ H_{16}$	Heptane		
		90 to 120.0	194 to 248	.....	.....		
Benzine (Octanes and nonanes)	.....	.....	.....	$C_8\ H_{18}$	Octane		
		120 to 150.0	248 to 302	.....	.....		
Kerosene	48	.....	.....	$C_9\ H_{20}$	Nonane	.....	
Vaseline	.....	.....	.....	.....	.....		
Paraffin	.....	.....	.....	.....	.....		

The commercial products of distillation of crude oil are given on the left in the table, together with their respective boiling temperatures and specific gravities. Some of the chemical constituents of these products, together with others forming the natural gas found in connection with these oils, are given on the right, also their boiling temperatures.

The table shows that such products as petroleum ether, naphtha and benzine overlap one another chemically and that refinery gasoline partakes of some of the constituents of rhigolene, petroleum ether and naphtha.

The natural-gas from oil wells from which the gasoline is made by condensation contains not only methane, propane and butane, all of which are gases at ordinary temperatures, but also vapors of some of the heavier constituents forming the oil. Since gasoline is composed of the lighter of these oil constituents, it is economically desirable to recover all the vapors, but in the process by which they are condensed some of the lighter gases are also liquefied. The more of these higher constituents the condensate contains, the more volatile it will be. This property of condensed gasoline is offset to some extent by "blending" with the heavier product, naphtha, which, as its boiling temperature indicates, is far less volatile than the other product.

Before methods of blending were developed the condensed gasoline was "weathered," by which process the more volatile constituents were allowed to evaporate to such an extent that the residual product could be used for the same purposes as the refinery gasoline and could be shipped with the same safety. In weathering, as much as 60 or 70 per cent. was sometimes allowed to go to waste.

The natural-gas and oil industries have always been notoriously wasteful. In some wells the value of gasoline recovered by present wasteful methods but formerly allowed to go to waste, may amount to 20 per cent. of that of the oil produced.

Tom Hunter, Hoisting Engineer

By W. O. Rogers

SYNOPSIS—We visit an iron-mine plant and find an interesting "stunt" that prevented the boiler-feed pump from becoming steam bound in case the firemen should open wide the feed valves and drain the open heater. Hunter and the chief engineer discuss the value of feed-water heaters, and I learn something of ore-skip hoists.

"Well, what is the program for today?" I asked Hunter the morning after our trip to the Woodward mines, as we ate our cold-storage, soft-boiled eggs for breakfast. "I'm game for another mine expedition if you are." "I'm fit as a fiddle," replied Hunter. "We'll take a run out to the Republic Iron & Steel Co. They are located on the same range of mountains not far from yesterday's tramping grounds." "I understand that these iron mines are mostly of the slope variety and are not deep as compared with coal mines," I remarked, as Hunter placed a plugged ten-cent piece on the table as a tip for the waiter who had asked for the breakfast order twice and had failed to get it right then. "There are some deep mines in Pennsylvania where we were last year. The deepest is the Brookside in Schuylkill County; it is 1854 ft. deep. The second deepest is the Auchincloss shaft at Nanticoke, with a depth of 1726 ft. That is where they wanted you to go down to see the gas burn around a safety lamp where you would have been about four miles from the shaft. You balked at that, excusing yourself by saying you had a wife and three small children awaiting your return. Gee! That was funny! Why, you would be as safe down there in the dark as you are riding around the country in a Pullman car. On this last run we had two brake beams come down while running at full speed, and both within twelve hours."

"Yes, I know, but it didn't seem safe, and over 1700 ft. underground is rather deep. 'It's a long way from Tipperary.' What are some of your other deep mines?"

"Well, there is the Dorrance shaft in Wilkes-Barre, 1127 ft. deep; the Kingston No. 2 near the city, which is 1147; the Inman shaft in Luzerne County, with a depth of 1146 ft.; and others between 1200 and 1300 ft. deep. Near Pittsville the Wardesville shaft is 1476 ft. deep, and the shaft having the least depth is the Riverside in Minton, which is 276 ft. deep. These are anthracite mines. The deepest bituminous shafts run from 300 to 697 ft., the latter being the depth of the Coleman shaft at St. Michael.

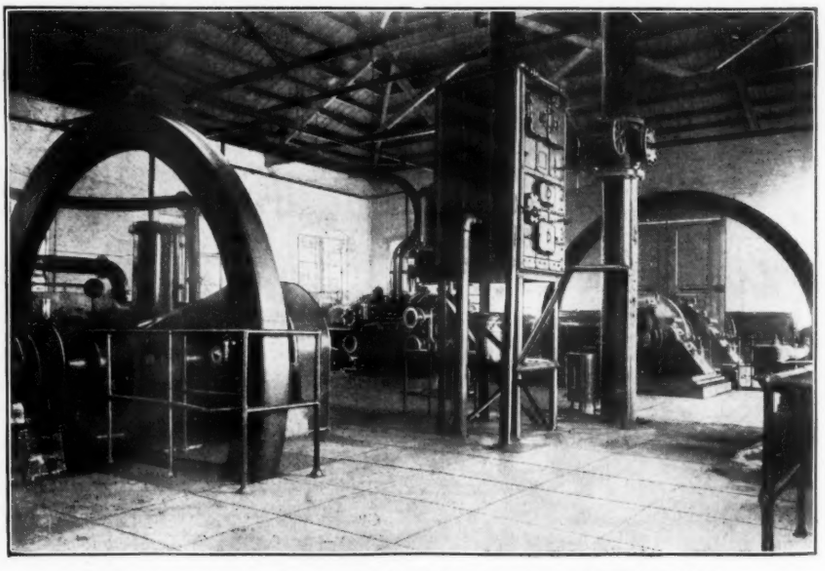


FIG. 1. COMPRESSOR ROOM AT THE REPUBLIC IRON AND STEEL CO.'S MINES

"In Germany the mines average about 1700 ft., the limits being 1000 and 2600 ft. So you see you wouldn't have reached the limit if you had gone down in the Auchincloss mine to see the gas burn."

We had, during this impartation of information, left the hotel and were well on our way to the mines. Arriving there, we introduced ourselves and were taken to the engine room, Fig. 1, where we found a cross-compound, two-stage air compressor, the steam valve being of the Corliss type. The machine had a capacity of 3000 cu.ft. of air per minute at 70 lb. pressure, running at 65 r.p.m. A similar unit with a capacity of 4000 cu.ft. per min. ran at the same speed. The cylinders were, for one machine, steam, 22&38x 42-in.; air, 35½&21½x42-in.; and for the other, steam, 25&40x48-in.; air, 37&22x48 in. The air pressure in the low-pressure cylinder of the two compressors was 23 lb. Steam was supplied from six 150-hp. return-tubular boilers. The chief engineer told us that at times the firemen caused trouble by allowing the water in the boilers to get low, say down to the bottom gage, and would then open the feed valves wide, which would allow the pump to empty the open heater and get steam bound, the water supply not coming to the heater as fast as the pump took it away. "But I got 'em now so they can't cut up their didos," laughed the Chief. "How is that?" asked Hunter. "Come over to the heater and I will show you;" which we did. While the Chief talked I made a sketch (Fig. 2). The heater rested on a frame of channel iron, with a shelf about midway between the floor and the bottom of the heater, which made a handy place for tools, etc.

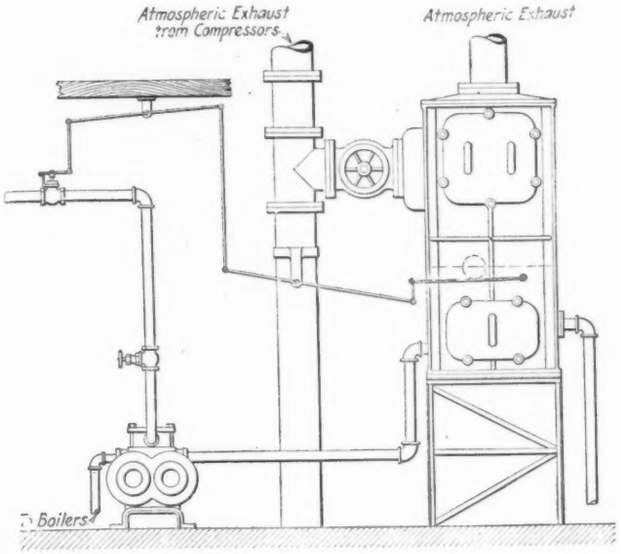


FIG. 2. ARRANGEMENT OF THE AUTOMATICALLY OPERATED PUMP STOP VALVE

"The pump," continued the Chief, "is fitted with a governor that is controlled by the pressure in the discharge pipe to the boilers, the same as used in general practice. But, you see, with that arrangement, when the firemen opened the feed valves at the boilers the pressure in the feed would be reduced and the governor would allow the pump to speed up and pump more water from the heater; then the pump would get steam-bound, knock and kick up generally.

"Well, to get the best of them fellers, I connected the float lever to a series of levers arranged to operate a quick opening and closing valve in the steam pipe leading to the feed pump. With this arrangement, when the water in the heater lowers, the float falls, and because of the way the levers are arranged the valve is partly closed. If the water gets low enough the steam supply to the pump is cut off, and the pump will stop until the water in the heater again reaches a stage where there is no danger of the pump becoming steam-bound and causing trouble."

"Quite an idea," said Hunter, "but what will happen if the water in the boilers is low and the auxiliary stop valve stops the pump?"

"Well, we had a little bother at first, but just as soon as the firemen found out that the pump would automatically stop when too much water was suddenly taken out of the heater, they carried the water at a more even height and consequently, the boiler room has not lost anything in economy."

"The way those firemen handled the water didn't do the boiler plant any good, that's a fact," commented Hunter. "Of course, the water should go to the boilers so hot that there will be no expansion strains produced, because of the entering water cooling the plates at one point, and other parts of the boiler remaining at a higher temperature.

"Letting a lot of comparatively cold water into a boiler all at once makes the coal that is burning in the furnace do the work in heating it, which should have been done by the exhaust steam in passing through the heater."

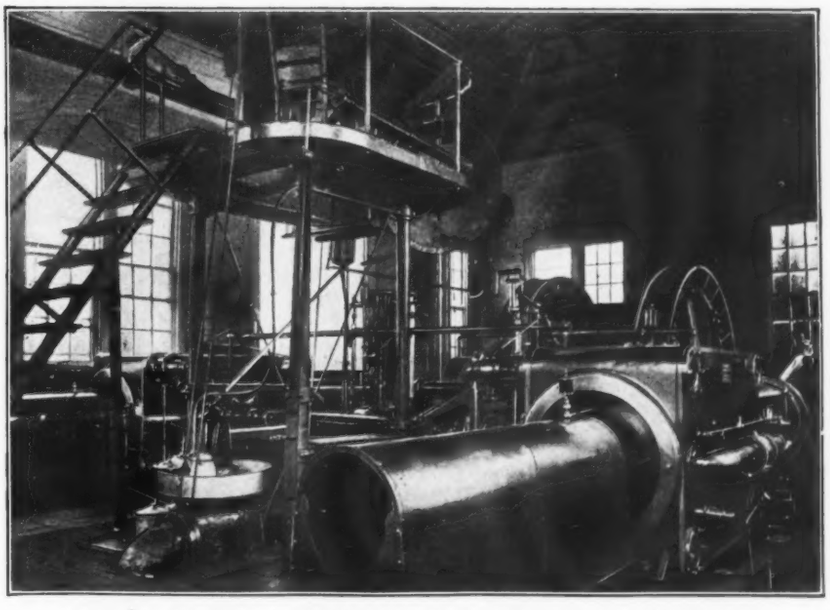


FIG. 3. HOISTING ENGINE AND ELEVATED OPERATING STAND

"About what saving does a good heater make in an ordinary plant, say one like this?" I asked, as we made our way to the hoisting-engine house.

"Well, say the intake water is 60 deg. and it is raised to 210 deg., which is done in many plants, with exhaust steam at atmospheric pressure, the saving in fuel should be about 15 per cent., which is 1 per cent. of the fuel for each 10-deg. rise in the temperature of the feed water; that holds good in almost any case."

"Well, I know this, the use of a heater increases the boiler capacity, and there is no mistake about that," interjected the engineer.

"There is no doubt of it," said Hunter, "the capacity of the boilers is increased about in proportion as the saving in coal. For instance, take a plant of half a dozen boilers and a good heater. They should make as much steam as seven boilers would without a heater, and with less fuel and fewer men."

"Yes, sir," exclaimed the Chief. "With average conditions such as we have here, 1 lb. of exhaust steam and 6 lb. of cold water will give me 7 lb. of hot feed water in that heater, while in some it would require 1⅙ lb. of exhaust at the least to get the water to the same temperature. We don't have any too much exhaust steam here, because the boilers supply steam to other units besides the compressor engines, and the way I have figured it out, with my heater I can get about 10 deg. higher temperature than I could if I were using some other types. I figure that the 10- to 15-deg. difference in feed-water temperature will amount to about 1 per cent. in the fuel bill.

"For instance, I have here a 900-hp. boiler plant, which I run but 10 hr. per day and 310 days per year. I use about 5400 tons of coal per year, which at $1.25 per ton would be $6,750, and 1 per cent. of that is $67.50 saved per year on the extra heat in the feed water. That may not mean much to some, but it is a mighty good illustration of what a little saving here and there might amount to in the ordinary steam plant."

"Right you are," responded Hunter, "and those little things amount to a lot more than most engineers realize."

We had by this time come to the hoisting-engine room where we found a 36x60-in. hoist equipped with a 10-ft. drum, which gave the hoisting cable a speed of 3500 ft. per min. with a load of 17 tons including the skip and ore.

The hoist engine is shown in Fig. 3. While Hunter was talking shop with the Chief, I "snapped" the skip just as it was about to mount the dumping track and also in the dumping position, Figs. 4 and 5.

The hoist engineer operates from a raised platform. The engine is equipped with a Corliss valve gear and the

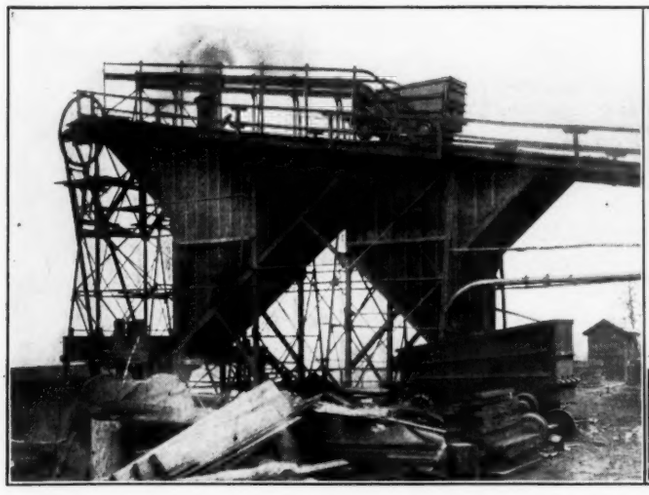


FIG. 4. SKIP WHEN ABOUT TO DUMP

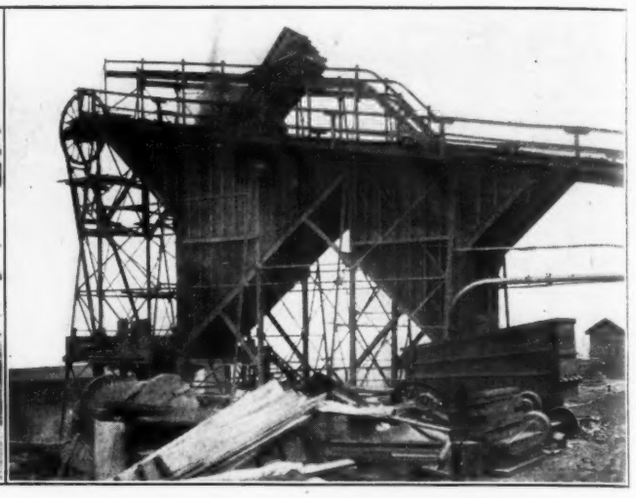


FIG. 5. SKIP IN ITS DUMPING POSITION

receiving motion and brakes are steam operated. I was interested in the skip operation. From the mine to the tipple the skip wheels run on the same rails. When the skip reaches the dumping track a smaller wheel on the outside of the two rear wheels rides on a track a little wider than the one running to the mine. This wider gage track is made in the shape of an ogee bend, with the horizontal track parallel with the one below and at a height such that as the rear track of the skip mounts it the ore is discharged from the end of the car to the crushing rolls, from where it is loaded into cars and taken to the iron furnaces.

The afternoon was well along by the time we finished our visit with the Chief, and so we decided to postpone further investigation until the next day.

New Compensator-Type Relay

A new type of circuit-opening inverse time-limit oil-dashpot relay has recently been developed by the General Electric Co. for use in conjunction with a low-voltage release for automatic, overload and low-voltage protection of alternating-current motors up to 2,500 volts and 300 amp. The relay is connected in series with the line, the low-voltage release being across one phase in the usual manner, with the low-voltage coil in series with the relay contacts.

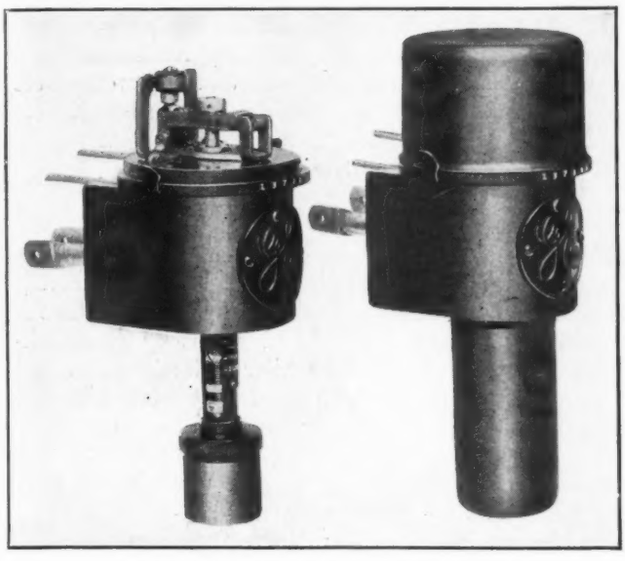
On overload greater than the current setting the relay, the relay contacts open-circuit the low-voltage release coil and the motor is cut out of circuit. If the voltage drops to a predetermined percentage of normal, the motor is also disconnected from the power supply.

This relay is employed mostly with motors using self-contained compensator control, but sometimes for switchboard service when both low-voltage and time-delay overload protection are required. Here series relays replace the secondary relays, current transformers and oil-switch tripping coils otherwise required.

Current calibration is from normal to twice normal, and the time adjustment from 10 sec. to 5 min. on 25 per cent. overload. The delay recommended, however, is

about 15 sec. at the starting current of the motor. This affords ample protection to the motor against damage from overload or single-phase operation, but prevents the circuit from being opened while the motor is starting.

The new relay is claimed to be a vast improvement over the one previously manufactured. The contact, dashpot and calibrating tube are inclosed by dust-proof stamped steel covers, and the current and time adjustments are accomplished outside of the dashpot simply with the aid of a screwdriver. The settings are constant, for an adjusting nut is locked in place after each setting is made.



RELAY WITH AND WITHOUT CASING

Initial Charge with Anhydrous

By H. G. Gibson

The usual method of charging an absorption plant for the first time is to inject a charge of aqua ammonia after the air has been withdrawn, and then increase the strength of the liquor as desired by adding anhydrous while the plant is working. It would be cheaper to buy all the ammonia in the anhydrous form and do the mixing in the plant, but ordinary tap water, undistilled, should never be used for this purpose. However, the cost of distilling water is trifling, and even if there is no apparatus for that purpose available, it is a simple matter to improvise one.

The necessary amount of distilled water should be obtained and a pipe run from this supply to the suction side of the ammonia pump; then run a line from the discharge side to the top of the cooler shell or, if that is not feasible, to some point in the weak liquor line between the generator and absorber. These temporary lines are not to interfere with the permanent suction and discharge lines attached to the pump, and each should have a valve with which to control the flow of water as conditions require.

By means of a vacuum pump or, if that is not available, the ammonia pump, take all the air possible out of the machine. Connect a drum of anhydrous and begin to expand it, at the same time pumping a little distilled water into the cooler or absorber, as the case may be. When the gage-glass shows liquor the water is pumped into the generator; continue this operation until the liquor in the generator is at the proper height.

The gravity of the mixture should be taken at intervals and the water supply regulated accordingly. A little practice will enable the operator to manipulate the various valves to best advantage. A small pipe-and-valve outlet taken from a tee in the discharge line from the pump to the generator will enable the gravity to be taken as often as desired. After enough liquor has been put in the generator to run the plant the temporary pipe connections may be dispensed with, and if the charge is not of the proper strength the required amount of anhydrous may be put in under working conditions.

In case the proper volume of liquor is obtained in the generator before enough anhydrous has been put in and before it is desired to start the plant, the injection of distilled water will have to be discontinued. The addition of more anhydrous may be continued by changing the temporary suction line to the ammonia pump, so that it will draw from the bottom of the generator instead of from the distilled-water supply. If this is done and it is found that the liquor in the generator is so strong that it will not readily absorb the anhydrous, it may be weakened by boiling the anhydrous over into the condenser, after which the operation may be resumed.

Mention has been made of two points of admission for the distilled water, either one of which may be used. As the anhydrous will have to be put in through the cooler, the best place for the introduction of the water is the top of the cooler. If the arrangement is thus made, a bypass should be provided from the purge line, where it leaves the bottom of the cooler, and the suction side of the ammonia pump. This would enable the mixture to go directly into the pump from the cooler instead of having to pass through the absorber. This bypass is not absolutely necessary, but its installation is desirable. If it is put in, a permanent job should be made of it, for it may often be used to pump out dead liquor from the cooler and thus save the time required to perform the same operation through the absorber.

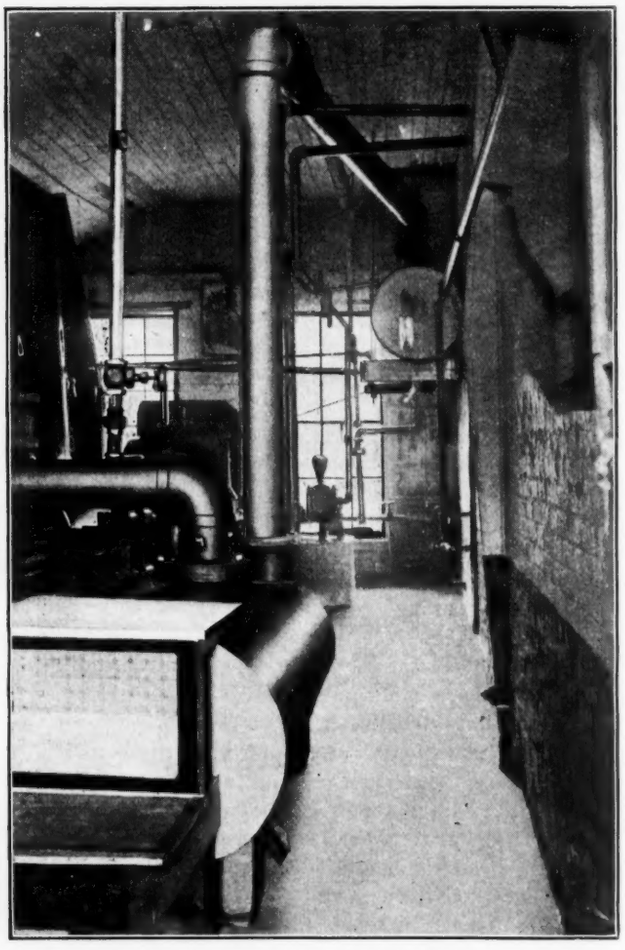
It will have been realized that where the foregoing plan is followed the cooler is temporarily made an absorber. If the usual absorber was fitted with a charging connection it would be the logical place to perform the mixing, for then the mixture could be pumped directly from the absorber without the necessity of getting it into the absorber from the cooler or of putting in a bypass line. However, it is much simpler to temporarily use the cooler for this purpose, and as the task is so seldom necessary, the installation of an extra charging set would be a waste of time and money.

Aqua ammonia (about 30% solution by weight) is quoted at 8¾c. per lb. and anhydrous at 25c. A 500-lb. drum of 30% aqua would come to $43.75 and contains 150 lb. of ammonia; 150 lb. of anhydrous at 25c. comes to $37.50; a difference of $6.25 on 150 lb. in favor of the anhydrous. Then again, the higher freight charge on aqua due to its greater weight would still further increase the relative cost of ammonia purchased in that form, unless the plant happened to be within short shipping distance of the consumer. Finally, when the prime charge is mixed in the plant the operator can be certain that only distilled water is put in. There is no infallible method, so far as I can find out, of accurately determining the purity of aqua ammonia.

Feed-Water Heaters in Tandem

The accompanying illustration shows the arrangement of a feed-water heating system that utilizes all of the exhaust steam from the engine with medium loads and most of it with a heavy load.

The main heater is placed alongside of the engine, with the exhaust pipe connecting at one end at the top. After passing through the heater, the steam not condensed goes through the vertical pipe at the other end to a second heater, which is supported on a platform at the end of the engine room and above the boiler-feed pump. The inlet pipe is connected at the top of the heater.



HEATERS CONNECTED IN TANDEM

The exhaust pipe extends from the other end out through the wall and up above the roof.

Although this is not the ideal heater arrangement, it illustrates how an engineer who is operating a plant with insufficient heater capacity can utilize the heat in the exhaust steam to better advantage by utilizing a second unit.

In this case the heater next to the engine raised the temperature of the feed water to about 120 deg. and a large volume of exhaust steam escaped to the atmosphere. By the addition of the second heater the temperature was increased to about 190 deg., the water flows to the boiler-feed pump by gravity.

The heaters are both made with inside return coils and are not of the most efficient design. The combination, however, gives good results, and but little exhaust steam escapes to the atmosphere.

JUST FOR FUN

Was His Own Head On or Off?

We were being bothered with oil collecting in the boilers and occasionally had to boil them out with caustic soda. On this particular occasion the boiling was being done under slight pressure. When my relief came on I told him what had been done and that the orders were to keep from 10 to 15 lb. pressure on this boiler. He absent-mindedly inquired, "Is the manhole plate on or off?"—George B. Longstreet, Medford, Mass.

Surely, Why Not Bright Lights?

The power station where I am employed is in the heart of the shingle-mill district, and my neighboring engineers, employed in the mills, frequently call in for a visit. Yesterday one called and inspecting the switchboard closely, he spotted the exciter voltmeter and asked if it was the one for the main generator. I told him that it was not and pointed out the machine voltmeter, registering 2,500 volts. "Gosh!" he exclaimed. "No wonder you fellows have fine bright lights! Why, I have only 115 volts and the mill hands are kicking something fierce!"—Ed. M. Keys, Jr., Montesano, Wash.

Pigheaded Persistence in an Error

The chief at a plant where I once worked wanted a 45-deg. elbow and, not having any in stock, undertook to make a substitute. He took a 90-deg. elbow and cut it in half, then tried to tap a thread inside and around the bend, but soon saw his mistake. He would not, however, give up his original idea, so filed down the end of the nipple and soldered the two together. His final fitting was one of about 65 deg. Needless to say, the fitting did not match and the connection did not hold the pressure.—H. W. Beard, Mendota, Calif.

A Narrow Escape—From the Ancients

As the engineer was filling the boiler preparatory to firing up, he noticed a colored attendant about the plant, gazing intently.

"What is on your mind, Sam?" inquired the engineer.

"I'se jist been reckoning, Cap'n, it shore would be turrble to be cooped in dat boiler now an' no way to get out," replied Sam.

"Well, Sam," said the engineer, "one time a man was accidentally shut in a boiler which was then filled with water to the second gage and a fire started in the furnace. After firing for some time and having raised the steam pressure to about 30 lb., the fireman noticed something moving up and down in the gage-glass in a violent manner. He called the engineer's attention to it and they found that it was a lead pencil. Immediately realizing it must be the missing man inside the boiler endeavoring to attract their attention, they hastily pulled the fire, drained out the water and knocked the manhead in. The man inside was released. That surely was a narrow escape, Sam."

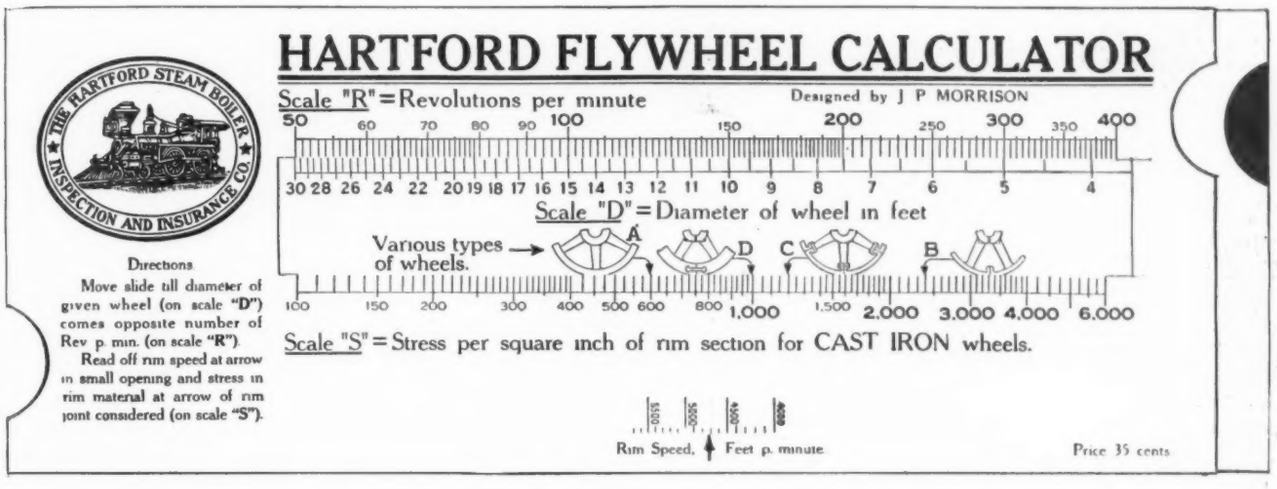
After gazing some more and thinking it all over seriously, the darky said, "Say, Cap'n. I done reckon dat you'se mighty careless how you all handles de truf."

No doubt others will think the darky's statement nearly correct.—C. E. Kling, Philadelphia, Penn.

Handy Flywheel Calculator

The flywheel calculator (shown in the illustration) that has been developed by the Hartford Steam Boiler Inspection and Insurance Co. for the use of its inspectors, was designed by J. P. Morrison, chief inspector at the St. Louis office, and should be of interest to engineers. It was primarily designed to determine the safe working speed for flywheels with various types of rim joints, but it can be used to determine pulley sizes required for obtaining desired speed of shafting and the velocity of belts, etc.

The caption under the lower scale—that is, "Scale *S* = stress per sq.in. of rim section for cast-iron wheels"—


HARTFORD FLYWHEEL CALCULATOR
THE HARTFORD STEAM BOILER INSPECTION AND INSURANCE CO.
Designed by J P MORRISON
Scale "R" = Revolutions per minute
50 60 70 80 90 100 150 200 250 300 350 400
30 28 26 24 22 20 19 18 17 16 15 14 13 12 11 10 9 8 7 6 5 4
Scale "D" = Diameter of wheel in feet
Various types of wheels.
A D C B
100 150 200 300 400 500 600 800 1,000 1,500 2,000 3,000 4,000 6,000
Scale "S" = Stress per square inch of rim section for CAST IRON wheels.
Directions
Move slide till diameter of given wheel (on scale "D") comes opposite number of Rev p. min. (on scale "R").
Read off rim speed at arrow in small opening and stress in rim material at arrow of rim joint considered (on scale "S").
Rim Speed, Feet p. minute
Price 35 cents

HANDY FLYWHEEL CALCULATOR

may be slightly misleading. It was found difficult to arrange for a proper description of this scale in a few words. It does not really represent the stress per inch of rim section, except for type *A* wheel, but it represents what would be the relative rim stresses in the other types of wheels, if the joint efficiencies usually found in wheels of those types were applied, or if the actual rim stress as given under type *A* were divided by the joint efficiencies applicable to the other types.

The scale as set in the illustration gives the rim stress per square inch section of type *A* at 4,712 ft. per min., which is that of a 15-ft. wheel at 100 r.p.m. or a 6-ft. wheel at 250 r.p.m. etc., as 600 lb. This divided by the relative efficiency of type *B* would equal 2,400 lb. per sq.in. In other words the value given in the central scale *S* is the "stress per square inch of rim section" only for a continuous ring of cast iron of uniform section, and this value is indicated by the arrow attached to the wheel segment marked *A*, the continuous rim of which is assumed to be the equivalent of such a ring. Suppose a stress of 600 lb. per sq.in. of section were allowable upon such a wheel; setting the arrow on *A* opposite 600 on the *S* scale would indicate on the little scale below that the rim speed was about 4,700 ft. per min., and the upper scale will give the corresponding number of revolutions per minute for a wheel of any diameter, or conversely the diameter of wheel which, run at a given number of revolutions, will produce the given rim speed and stress per square inch of rim section.

When the rim is weakened by a joint as in the segment marked *D* or made up like that marked *C*, or is loaded with a flanged joint midway between the arms as in that marked *B*, a considerably lower rim speed is allowed. The stress varies as the square of the velocity or the velocity as the square root of the stress. A wheel of the type *B* would be allowed only one-half the rim speed of one of the type *A*. It is therefore assumed that for the wheel of the type *B* the stress per square inch of section is four times that in the *A* type, which would require the wheel to run at one-half the speed to produce the same rim stress as in type *A*. Notice that when the arrow *A* is on 600 that at *B* is on 2,400 (four times as much) and that if *B* is moved to 600 the rim speed indicated upon the small lower scale, the r.p.m. for a given diameter, or the diameter for a given rotative speed upon the upper scale, will be halved.

The scale is a handy "pocket size" being 6½ in. long by 2½ in. wide.

Loss of Combustible in Ash

Did you ever look at the ash pile after a rain and notice the amount of coal staring you in the face? You do not notice it when it is pulled out of the ashpit, but it is there just the same. A certain percentage of fuel loss in the ashes is unavoidable, the amount depending upon the size and kind of fuel and the condition of the grate bars, but under good operating conditions this loss should not exceed 5 per cent. of the total heat value of the fuel.

Most engineers would say that a plant operating with 8 to 10 per cent. of combustible in the ash was doing as well as the average, and but few realize the yearly loss in dollars or the saving that can be made by reducing the loss of fuel in the ashes.

The accompanying reports, which appeared in the September number of the *A. G. E. Bulletin* of the American Gas and Electric Co., of New York, give the loss due to combustible in ash at their various power plants, the figures being based on laboratory tests.

It should be stated that the coal burned in the several plants was not of the same grade and that the combustible content of the coal burned was higher in one plant than in another. The figures given in the estimated yearly loss and the estimated yearly saving if all of the plants

reduced the combustible in the ashes to that of the plant designated as a "standard" do not indicate that the plants showing the higher losses can approach the showing of the "standard" plant. The figures are valuable

LOSS DUE TO COMBUSTIBLE IN ASH

Coal loss due to combustible in ash, as shown by analysis of samples of coal and ash from the several plants at laboratory of the Scranton Electric Co., Scranton, Penn.

Report for Week Ending July 31, 1915

Plant	Per Cent. Combustible Loss	Equivalent Coal Loss, Per Cent.	Estimated Yearly Loss	Estimated Yearly Saving if all = Atlantic City
Atlantic City	3.38	3.85	$4,535	.....
Wheeling	4.21	4.74	2,268	$426
Fremont	4.99	6.32	1,287	503
Tiffin	4.48	5.39	2,095	633
Scranton	7.50	9.89	6,560	4,005
Total			$16,745	$5,567

Report for Week Ending Aug. 7, 1915

Plant	Per Cent. Combustible Loss	Equivalent Coal Loss Per Cent.	Estimated Yearly Loss	Estimated Yearly Saving if all = Newark
Atlantic City	7.45	8.08	$6,850	$2,997
Scranton	7.15	9.33	6,194	4,208
Wheeling	4.30	4.71	2,257	1,419
Tiffin	9.05	10.68	4,020	2,997
Newark	2.47	2.97	482	.....
Total			$19,803	$11,621

Report for Week Ending Aug. 14, 1915

Plant	Per Cent. Combustible Loss	Equivalent Coal Loss, Per Cent.	Estimated Yearly Loss	Estimated Yearly Saving if all = Newark
Scranton	8.19	10.72	$7,105	$6,149
Atlantic City	5.70	6.46	5,480	4,248
Tiffin	6.41	7.89	3,062	2,510
Newark	1 29	1.45	236	.....
Fremont	3.82	4.45	907	611
Wheeling	5.00	5.57	2,665	1,970
Muncie	6.74	8.34	8,180	6,758
Marion	12.43	16.82	5,370	4,905
Total			$33,005	$27,151

Report for Week Ending Aug. 21, 1915

Plant	Per Cent. Combustible Loss	Equivalent Coal Loss, Per Cent.	Estimated Yearly Loss	Estimated Yearly Saving if all = Newark
Atlantic City	4.43	4.94	$4,220	$1,965
Muncie	10.56	13.76	13,390	10,820
Scranton	8.14	10.71	7,120	5,236
Wheeling	5.36	6.10	3,042	1,715
Tiffin	7.71	9.25	3,760	2,742
Newark	2 32	2.64	433	.....
Fremont	6.99	8.54	1,718	1,187
Rockford	9.73	13.05	11,680	9,317
Total			$45,363	$32,982

to the engineer in showing the saving that can be made by reducing the loss of combustible in the ashes, and although it would be impossible to reduce this loss to the "standard"-plant basis, because of the character of the coal, type of furnace, etc., the figures show what the saving would be if it were possible to do so.

Unusual Water-Supply for Power Plant

By F. B. Hays

A power plant in a city was so located that its water-supply meant a large expenditure of money each month. To save this expense an underground water-works was installed, which had its source of supply in a river half a mile distant. The general arrangement of the system and its more important details are shown in Figs. 1 and 2. The waste water from the plant is discharged into the city sewerage system.

The object in locating the plant so far from a source of water-supply was to get as near the center of the

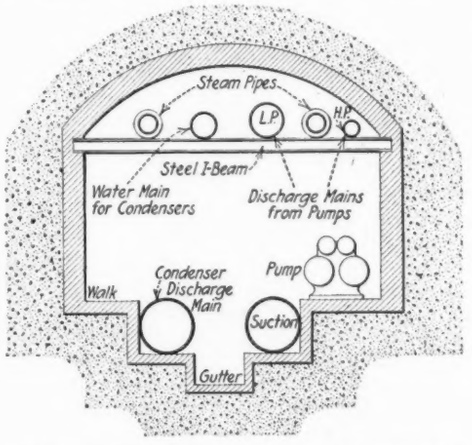


FIG. 2. CROSS-SECTION OF THE TUNNEL

city as possible and to locate on the main line of a railroad where the cheapest ground was available. The system has been in operation for a number of years and has given satisfaction.

San Francisco's Auxiliary Pumping Plants

Contrary to popular opinion, the high-pressure fire protection of San Francisco is not a salt-water, but a fresh-water, system. The Twin Peaks reservoir, which has a holding capacity of 10,000,000 gal., furnishes a maximum pressure in the distributing mains in the business portion of the city of from 300 to 350 lb. As an auxiliary in case of failure of the main supply, two pumping stations have been installed in connection with the system—one at Second and Townsend St. and the other at the foot of Van Ness Ave.

These stations pump water direct from the bay into the mains in cases of emergency, and the Townsend St. station may be used to supply pressure in the manufacturing district adjacent to the station, independently of the main supply. In this station are four steam turbo multi-stage pumps, each capable of furnishing 3,000 gal. per min. at a pressure of 350 lb.

No salt water is allowed to remain in the pumps or mains at any time. After use, the pumps are flushed with fresh water, and by the opening of a valve the mains are bled until the salt water has run out. The piping and

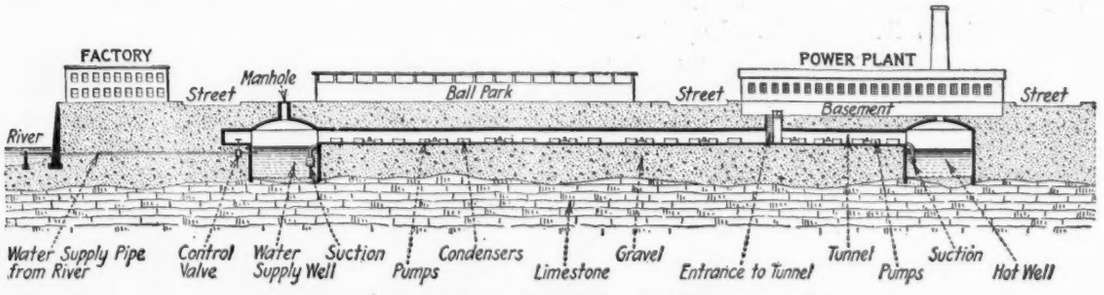


FIG. 1. ELEVATION OF THE WATER-SUPPLY SYSTEM

distributing system were dipped in coal tar prior to installation. The interiors of the pumps, which are of cast iron, have a coating varying from $\frac{1}{16}$ to $\frac{1}{8}$ in. thick of bitumastic enamel, and this has proved very satisfactory in use, no trouble from corrosion having been experienced. The runners and diffusing rings are of phosphor bronze.

The pumps are operated at least once a month for drill purposes. The boilers are oil-fired and operate at a normal pressure of 190 lb. A stand-by boiler is used. Whenever the pressure in the boilers runs down to 125 lb., the burners are lighted and the pressure run up again. This is necessary about every six hours.

Power Distribution on Penn. R. R. at Philadelphia

SYNOPSIS—An outline of the power distribution system on the electrified suburban service of the Pennsylvania R.R. at Philadelphia.

The electrification of the suburban service of the Pennsylvania Railroad between the Broad Street station and Paoli is the first electrification undertaken by that company in the vicinity of Philadelphia, and its primary purpose is to increase the capacity of the Broad Street station and thus relieve congestion at the terminal.

Analysis of the service conditions and cost estimates covering all available electric systems led to the conclusion that one using a high-voltage overhead contact and one which eliminates moving machinery in the substations was most suitable and economical from the standpoint of first and of operating costs. In arriving at this conclusion consideration was given to the possible long-distance operation over the entire division rather than to the requirements for the present short suburban electric service. Accordingly, it was decided to use 11,000-volt single-phase 25-cycle power applied directly to the trains from an overhead catenary trolley system.

Power for traction is supplied by the Philadelphia Electric Co. and is delivered to the railroad company at

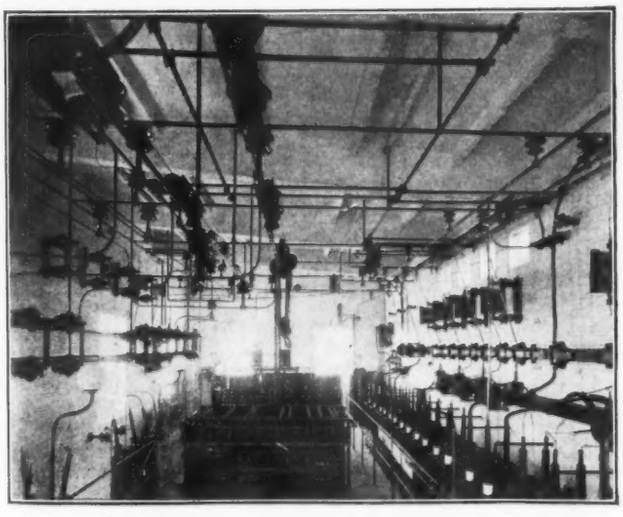


FIG. 1. SUBSTATION INTERIOR, SHOWING OPEN BUS STRUCTURE

a substation on the westerly banks of the Schuylkill River, by means of an armored submarine cable. At present the service is on one phase only of the power company's three-phase generating system, but it is planned to supply future requirements from the remaining phases. Provision has been made by the Philadelphia Electric Co. to balance this single-phase load, as well as to correct for the relatively low power factor, in order that the full three-phase capacity of their generating units may be available.

The power is delivered at 13,200 volts, to the main substation where it is stepped up to 44,000 volts and transmitted to three step-down substations. The main substation at Arsenal Bridge contains three 5,000-kv.-a. step-up transformers and each of the three step-down substations contains two 2,000-kv.-a. transformers which reduce the voltage from 44,000 to 11,000 volts for use on the trolley wire. Each of these substations is a fire-

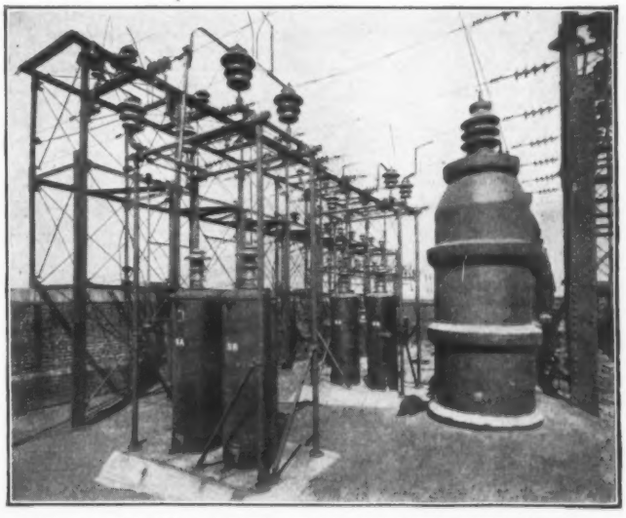


FIG. 2. SECTIONALIZING SWITCHES AND LIGHTNING ARRESTERS

proof brick building housing the transformers on the first floor, the busbars and switching equipment on the second floor and the lightning arresters and high-tension feeder-sectionalizing switches on the roof. Space is provided for doubling the capacity of each substation. Open busses mounted on insulators carried by pipe framework are used throughout, and practically all the power wiring is bare, copper tubing or solid wire being employed.

The transformers are all of the 25-cycle single-phase oil-insulated water-cooled type, having the cases mounted on wheeled trucks, and are equipped with thermostats that operate an alarm bell in case of high temperature. Oil-filtering and drying apparatus is located in each substation, and a tank is provided into which the oil may be drawn from the transformers.

The neutral points of the high-tension windings of the step-up transformers are grounded through grid resistances, thus limiting the potential to ground from either side of the 44,000-volt transmission line to 22,000 volts. The lightning arresters on all the 44,000- and the

11,000-volt lines are of the electrolytic type, and the circuit-breakers of the 11,000- and 13,000-volt circuits are of the oil type, those on the 11,000-volt trolley circuits being two-pole, with a reactor connected across one pole.

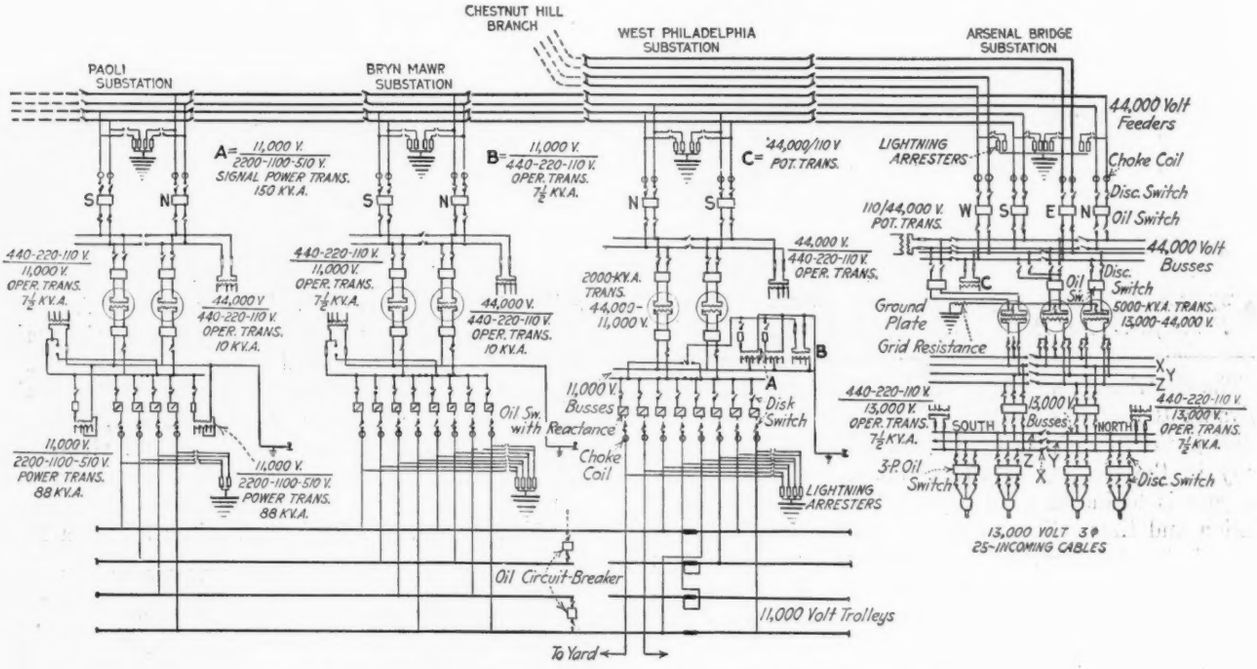


FIG. 3. DIAGRAM OF THE POWER-DISTRIBUTION SYSTEM

The circuit-breakers are automatic and remote-controlled. Power for the opening and closing of the oil circuit-breakers is obtained from the 44,000- or the 11,000-volt busbars, through small transformers stepping down to 420 and 220 volts. Sixty-cycle power is also provided in all the substations, to trip these circuit-breakers in case of failure of the 25-cycle power.

Except in the West Philadelphia substation, where the power director is located, there are no attendants. Switchboards with the necessary instruments, controllers and indicating lamps are provided in signal towers near the three unattended substations. These boards are connected with the boards in the substations through a control cable and the opening and closing of circuit-breakers are accomplished by the tower man. Telephones are provided in all the substations and in the signal towers controlling them, so that the power director is in constant touch with all substations and tower men.

In order to minimize the inductive effect of the traction currents on adjacent telephone and telegraph wires, a special system has been installed, consisting of a series of booster transformers, placed about a mile apart. These are mounted on the signal bridges and the current in each of the trolleys is constrained to pass through a primary coil of the transformer, thus producing an equal current in the secondary coil, the ratio of the transformer windings being approximately one. The secondary coils of these transformers are connected across insulated joints in the track, consequently as all trolley current must flow through the primary coil, the secondary coil insures that all the returning traction current at this particular point of the circuit at least, must flow through the rails. In this manner the traction current is prevented from taking ground returns parallel with the track, or if there is a tendency between these booster transformers for the traction current to seek a ground return, this tendency is corrected at the next booster transformer.

Each car contains two 225-hp. single-phase 25-cycle, commutator-type motors, which are connected in series and are started and operated up to approximately 15 mi. per hr. as repulsion motors.

Gibbs & Hill, of New York, acted as consulting and designing engineers on the work and the substation and car equipment was furnished by the Westinghouse Electric and Manufacturing Co.

The Corrosion of Gun Metal

The fourth report of the Institute of Metals Committee, on the subject of the microchemistry of corrosion, was made by Cecil H. Desch and Henry Hyman. The three previous reports have dealt with the alloys of copper and zinc. In this study, the tin bronzes, alloys of copper and tin containing a small percentage of zinc, were selected for investigation. The metal was attacked by electroiysis, and the general results may be summed up as follows:

Total corrosion is much less than that of brasses under similar conditions. The protective effect of tin is strongly marked, a white adherent film of basic salts being formed after a short time and exerting a mechanical influence in the direction of checking corrosion. An 84:14:2 alloy is much less corroded than the 88:10:2. Annealing the former, thus producing a coarser grain, has the effect of increasing the corrosion.

Cylinder Oil and Boiled Linseed Oil equal parts are said to have given good results on a hot crankpin in one instance.

Acid in Oil will cause corrosion when in contact with moisture. The presence of sulphuric acid may be determined by a solution of barium chloride in distilled water, a few drops of which put into the oil after it has been thinned with an equal amount of gasoline or benzine will cause a whitish precipitate if acid is present. In the absence of moisture the acid is not so active and may escape notice.

Welding Broken Machine Parts

SYNOPSIS—By the oxyacetylene process broken machinery is welded nowadays that would have been scrapped as beyond repair a few years ago.

Practically every manufacturing company has somewhere about the yard what is commonly called a scrap heap, where a miscellaneous collection of broken machines may be found, because there has been no attempt made to repair them by modern means. There was a time when there was nothing else to do, because in many instances the repair would have been impracticable either because of the expense or because the work would prove insufficiently strong to withstand the strain put upon the parts.

With the coming into use of the acetylene-gas welding apparatus a few years ago the scrap heaps grew smaller and many surprising repairs have been made by using this process to repair broken machinery. The accompanying illustrations show some of the jobs done with the Milburn oxyacetylene apparatus and are representative of the work that is being done with broken machinery parts.

Fig. 1 shows the main base casting of a gasoline engine that was broken in two, was welded as in Fig. 2 and restored to service practically as good as new. This work was accomplished in less than 8 hr.

Figs. 3 and 4 show the welding of a special gear wheel that was badly broken and looked like a total wreck. On account of its special construction and separate center, it was difficult to replace this gear, and a welding job was determined upon. Four welds were made in the rim, one spoke was welded on and the end of another one was built up where it had been broken out. This job was finished within 10 hr., saving 10 days or more in securing a new gear.

Fig. 5 shows a heavy casting of a rock-drill sharpener. The head was broken off, leaving two fractures, each 5x7 in. cross-section. The work was completed in about 13 hr., which included preheating in a charcoal furnace. The piece was restored to service as good as new and has stood up under heavy strain.

A large belt pulley had two large pieces broken out of the rim by a piece of flying rock. To repair this, three welds were necessary (Fig. 6), each being about 14 in.

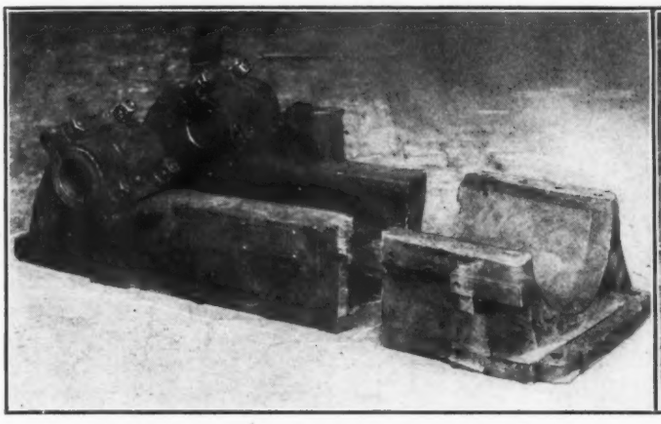


FIG. 1. THE BROKEN ENGINE FRAME

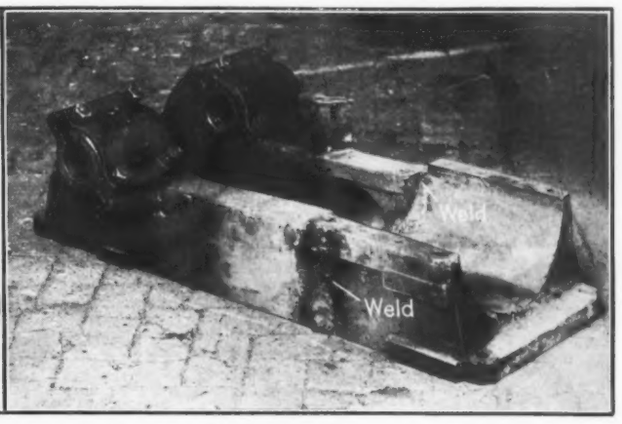


FIG. 2. ENGINE FRAME AFTER WELDING

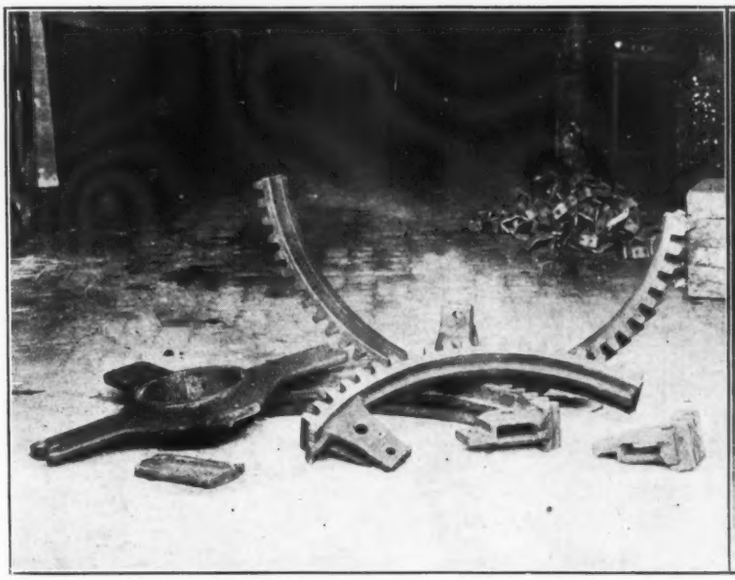


FIG. 3. GEAR WHEEL BROKEN INTO SMALL PIECES

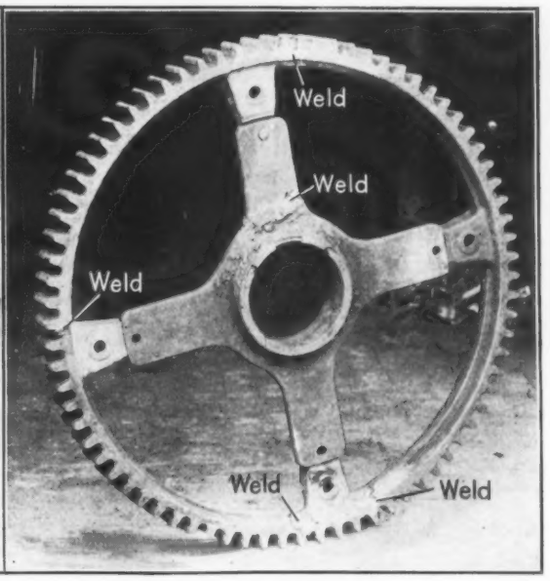


FIG. 4. GEAR WHEEL SHOWING WELDS

long in metal and about 1 in. thick. This repair was accomplished with a delay of only about 10 hr., and the wheel was restored to service smooth and true and as strong as new. The delay was slight and the cost low.

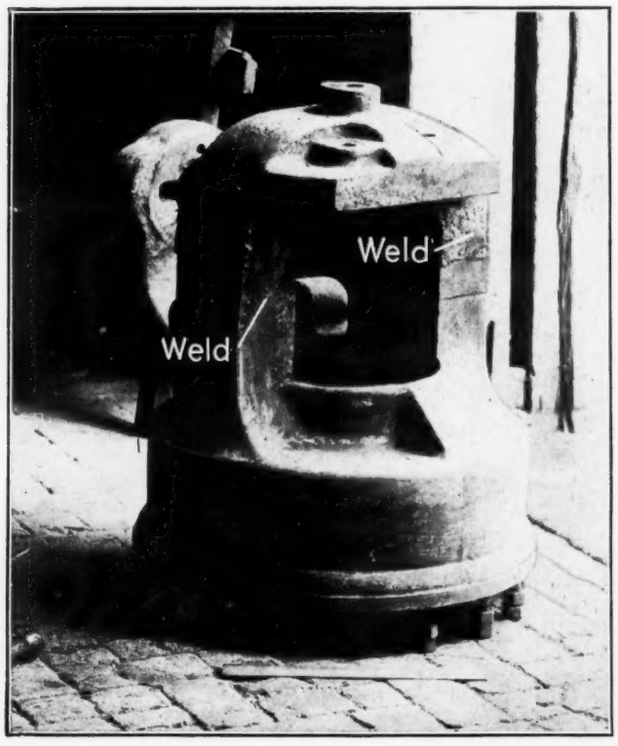


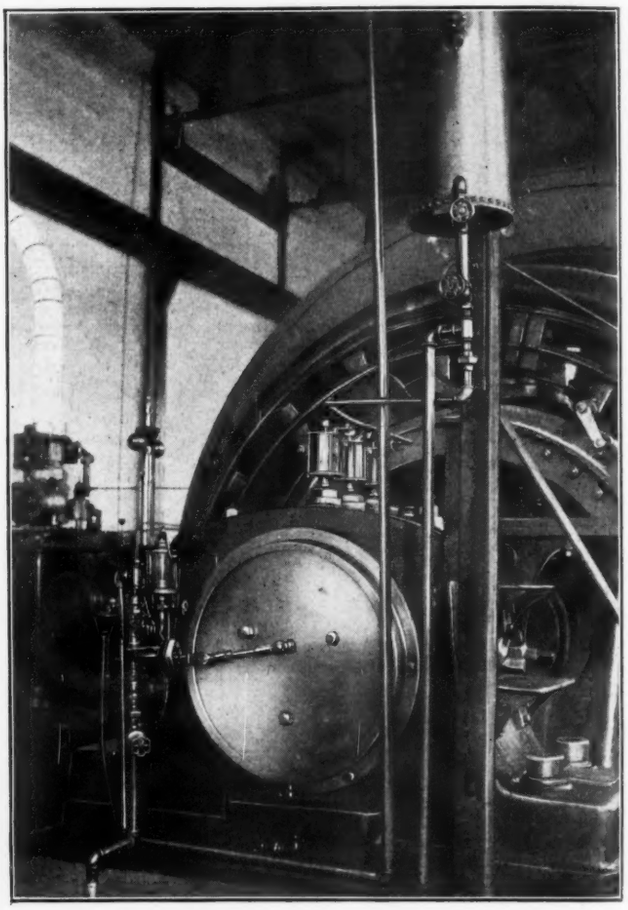
FIG. 5. HEAVY CASTING WELDED IN TWO PLACES

FIG. 6. BROKEN BELT PULLEY AFTER THREE WELDS

We are indebted to the Alexander Milburn Co., 1426 West Baltimore St., Baltimore, Md., for illustrations and data.

Bearing Lubricating System

An interesting method of lubricating the outboard bearing of a 32x48-in. Corliss engine is illustrated herewith. The pump barrel is made with a tee-like connection, the plunger working in the horizontal cylinder. The bottom and top outlets of the pump body are fitted each with a



OIL PUMP DRIVEN FROM CRANKSHAFT

check valve. The bottom one is piped to a filter in the basement, and the top connection runs to an oil tank supported by an upright pole.

From the bottom of this tank, connection is made to a small manifold that runs lengthwise of the bearings and from which small pipes lead to the oil cups, of which there are three. The oil, after passing through the bearing, is drained to the filter in the basement. The cups are made oiltight, each having a regulating needle valve by which the feed is adjusted. The pump plunger receives its motion from a crank-rod actuated by a pin set in the end of the engine crankshaft.

The Specific Gravity, Viscosity and Flashpoint of various oils are approximately as follows:

Kind	Sp.Gr.	Viscosity at 100 Deg. F.	Flash-point
Light mineral	0.860 to 0.875	27 to 35	340 to 360
Medium mineral	0.875 to 0.890	44 to 47	380 to 400
Heavy mineral	0.890 to 0.910	47 to 86	400 to 440
Cylinder mineral	0.900 to 0.905	350 to 900	500 to 550
Sperm	0.875 to 0.885	35	
Lard	0.910 to 0.920	60	
Tallow	0.910 to 0.915	65	475 to 500
Rape	0.910 to 0.915	55	
Olive	0.915 to 0.920	60	
Castor	0.960 to 0.970	270	
Cotton and corn	0.920 to 0.930	55	600
Rosin	0.960 to 1.000	50	

Editorials

Meeting Progress Conditions

One of the greatest things in human life is the ability to adapt oneself to his environment. The mammoth, dinosaurus tyrannosaurus and some more of those things we see in museums could not adapt themselves to theirs and they are extinct.

We frequently hear the cry that the great bulk of engineers are behind time, that they are asleep at the roadside as it were, while progress marches by, leaving them to become extinct. The turbine is often referred to as the whole parade in this progress procession, and the mourners weep because engineers are not keeping pace with it.

There are a great many thousand horsepower in turbines in service now. There was practically none fifteen years ago. Those running are being managed well, and the ever-increasing number going into service find competent operators.

Let the howlers cease. One trade or profession or sect does not get so far ahead but what the rest of us can keep pace.

Still in the Ring

Little evidence that the isolated plant is on the decline at Boston, appears in the latest report of the city Wire Department, covering the fiscal year 1914-15. At the close of the year there were no less than 323 isolated plants in service at Boston, embracing 73,579 hp. in boilers, 54,920 hp. in engines and 30,377 kw. in electric generators. From the busses of these plants 2,290 electric motors were receiving energy, the capacity of these totaling 19,827 hp., or an average of 8.65 hp. per motor. Three years ago the boiler capacity of the 324 isolated plants then listed was 70,998 hp., the engine rating was 50,736 hp., and the generators totaled 27,402 kw. Then, too, there were only 1,815 motors rated at 17,576 hp. in isolated-plant service; so that, barring a reduction of one in the number of plants, there has been a substantial development all along the line, and this in spite of vigorous campaigning on the part of the local central station.

The average engine rating of these plants is only about 174 hp. each, and the published boiler capacity greatly exceeds the engine ratings. This is to be expected in view of the prominence of the heating load. It is gratifying that the large power business of the isolated plants is well maintained, the average size of motor on the Edison lines being but 4.75 hp., as reported. In other words, the average motor operated by isolated plants is nearly twice as large as on the central-station system, indicating the peculiarly valuable characteristics of central-station service for the small consumer.

There are, of course, economic situations where the purchase of power from the central station is clearly the right thing; and, as has been pointed out before, what we are all after is the lowest-cost service consistent with first-class operation. There are nearly eight times as many motors on central-station service in the Boston district as there are on Boston isolated-plant circuits, but the magnitude of isolated-plant development, its sustained growth from year to year and its very considerable importance as compared with other power developments in the community should hearten the operating engineer, who is sometimes led to fear the central-station advance by the success that these companies are attaining. The fact that the number of plants privately operated has held so closely to the line for the past three years in a city like Boston is the best possible indication that the problems of power production and distribution, including heating, are varied and individual, and that cut-and-dried methods of procedure have little place in the expenditure of hard-earned money for plant output.

Laying Up Idle Boilers

Boilers tend to corrode when idle. How best to prepare them for long periods of disuse has been a much disputed question. Mention of this subject, together with directions for laying up such boilers, in C. E. Stromeyer's recent report to the Manchester Steam Users' Association will therefore be appreciated by many who are not familiar with the procedure he recommends.

The boiler, he says, should be emptied and thoroughly dried and trays of unslaked lime put inside, after which the boilers should be hermetically sealed. Unslaked lime is an effective absorbent of moisture, so that any that remains after the drying-out process will be taken up by the lime. Deprived of moisture, any chloride of magnesia or other remaining salts will be quite harmless. Also lime, unslaked, will absorb what carbonic and sulphurous acids may be present.

Notice that a high degree of dryness is essential to the success of this method, which, we believe, is as good as any and better than most. So stop valves that do not leak are important. Unfortunately, it has been our experience and observation that, although tight, if properly closed and cared for, stop valves are frequently allowed to leak unnecessarily.

Turning Cold Water into an Overheated Boiler

To many engineers the advice to turn comparatively cold feed water on boiler plates when they have been overheated because of low water, as given elsewhere in this issue, will be received with somewhat of a shock. It is contrary to the universal admonition of the dangers of not immediately drawing the fire or covering it with wet ashes and opening the fire-doors—all for the purpose of gradually cooling the overheated plate or plates to avoid stresses so serious as to probably bring about an explosion.

One can find good authority today to uphold that admonition. In the recent preliminary draft, third print-

ing, page 253, of the Boiler Code Committee's report to the American Society of Mechanical Engineers, one will find the time-honored advice. In the final report, however, the section relative to operation and care of boilers has been omitted, and of course the advice does not appear.

The Manchester Steam Users' Association, one of the oldest organizations devoted to safety in boiler construction and operation, in 1889 made experiments on a Lancashire boiler to determine what damage, if any, would occur as the direct result of turning feed water into an overheated boiler. The results of these experiments are not generally known among operating engineers of the United States. Briefly, no damage at all followed the introduction of water into the boiler. And C. E. Stromeyer, the present chief engineer of the association and a recognized first-class boiler expert, advises, in this year's report to the association, that no hesitancy be observed in letting feed water into an overheated boiler for the purpose of quickly cooling it.

So, after all, Mr. Blackburn, the author of the article referred to in this issue, has "a leg to stand on."

The Privilege of Decision

In many private plants the owners tell the newly employed chief engineer that they wish to rely upon his knowledge and judgment to a degree that will release them from all detail; whereat said engineer rejoices and is exceeding glad. In due course a question of policy arises involving the expenditure of several thousand dollars for new condensing apparatus; the engineer's advice asked and then disregarded. The installation is made against the engineer's trained judgment and naturally enough he resents it. "Where," he thinks, "is my standing as chief of this outfit, and where is the right of decision? Shall I quit, after the fashion of cabinets abroad whose policies are not sustained by the controlling vote?"

By no means should a man resign because of one such experience—perhaps not after several. Hard as it is to see a technical misjudgment put into execution, the engineer must appreciate, when looking at the matter dispassionately, that the man who puts his money into the plant has the undisputed right to equip it as he sees fit. It is up to the engineer to protset himself by recording his own views, privately at least, where he can refresh his memory later on with the facts and recommendations as he saw them, if the results of operation do not bear out the owner's expectations and perhaps tend to make the engineer the "goat."

Accepting the situation at its face value, the proper course, then, is to continue exercising the best possible technical judgment in the operation of the plant, making no recriminations against the owner among one's fellow-engineers or subordinates, but striving all the harder to make successful recommendations the next time a question of policy arises. The engineer who is sure of his position in an important problem of this kind, who cultivates the art of putting his opinions and data to his employer in a logical and convincing way and who follows this plan, does his whole duty in the premises. In the long run he is pretty likely to earn the privilege of decision, which is the expression of the owner's confidence in his professional ability; and although at times he may be overruled, he can accept the adverse decision philosphically and if events prove that he was right, can congratulate himself on his foresight. And if subsequent operations show that the engineer was mistaken, hecan learn why and avoid repeating the error in his later practice.

The man who is never overruled is a rare phenomenon, and the old motto of being sure one is right and then going ahead represents a pretty high ideal. To exercise sound judgment within the limitations of the available data is about the best most of us can accomplish, and the more this is done the greater progress we make toward superlative proficiency.

Standards and Working Conditions in Analysis

Standard methods of testing are convenient and form the most accurate means of judging the comparative merits of the materials tested. They should, however, be under conditions duplicating the actual commercial or industrial ones and not under special conditions established in the laboratory for convenience.

An interesting case in point came up recently in a large refrigerating plant in Boston. Two lubricating oils of about the same price were applied to the cylinders and valves of a large ammonia compressor in which the temperature was above four hundred degrees. One oil would not prevent the valves from dragging, the other gave no such trouble. Analyses of both oils in the laboratory showed them to be about identical. Naturally speculation as to where and how the difference existed was in order, and quite naturally the losing salesman was anxious to determine the cause—so more analyses were made only to verify the former ones.

The user of the oil suggested that inasmuch as the viscosity and other properties of the oils were taken at one hundred degrees, while the working temperature was much greater, that might explain why a difference in results existed. To obtain a viscosimeter that would handle oil at the cylinder temperature was possible only on a special order, as none ordinarily are made. Up to a temperature of over three hundred both oils compared favorably, but at comparatively little higher temperature one ran more than forty points better than the other.

Such experiences should convince us that if tests and analyses of at least some materials are to be dependable, they must approach as nearly as possible the working conditions.

The boss may not know as much about the engine room as you do, but if he foots the bills and pays your wages he is entitled to a look-in and a say-so now and then.

We wonder why illustrators so often picture the engineer as being square-jawed and husky. Probably they assume that because he handles the power plant he must be powerful.

To enable labor-troubled Great Britain to conserve its much-needed fuel supply, it is respectfully suggested that the services of Chairman Walsh of the Federal Commission on Industrial Relations be tendered that country of thermal-unit unrest. As an "industrial heat" provider, Walsh is the carboniferous It!

Correspondence

Engine Specifications

In these days of modern equipment and high efficiency the work of selecting and purchasing the proper power-house machinery is often placed in the hands of a consulting engineer, rather than given to the engineer who will operate the plant. The consulting engineer is, as a rule, better equipped to make the selection of the best type of engine to use under the existing conditions, on account of his wide experience with many plants, but there are numerous points that he might overlook that would appeal to the plant engineer, but which might cause him to overlook some of the more important points of design.

There are so many makes that it is possible to select one or more stock engines that will fill the most exacting requirements as to economy, speed, steam pressure and cost, and that can be bought for less money than one built to fit the whims of someone that wants to experiment in engine design at the expense of someone else.

The details of the engine, however, should be closely studied in making the selection. Most builders furnish specifications of their engines in detail, but sometimes verbal promises that may not be mentioned in the contract or materialize in the engine are made by an overzealous salesman.

Engine valves are probably steam-tight when the engine is new, but suppose that it is used where the steam is wet and the boilers foam over occasionally or where the other conditions are not conducive to keeping them tight, what assurance, borne out by tests in actual practice under similar conditions, is there to prove that any particular valve will be tight at the end of one or two years? What provision, if any, is made to keep it tight? The steam consumption may be low when the acceptance tests are made, and yet if an efficiency test were made at the end of one or two years it might show a considerable increase in the steam consumption per horsepower-hour.

When the engine is directly connected to a generator, it is impossible to raise the shaft more than ¼ to ⅜ in. without raising the generator. It is sometimes necessary to remove the bottom shell of the main bearing, which should be of such design that it can be easily taken out if the shaft is raised ¼ in. In some engines this is possible, in others it is not. This feature may be of assistance in keeping the engine in condition. The four-piece bearing is preferable to the two- or three-piece bearing used in some engines, because it admits of better adjustment and is more substantial. In a large engine it would be of advantage to have an adjusting wedge under the bottom section of the main bearing to assist in leveling the shaft.

Another convenience that appeals to the engineer and assists him in keeping the engine in condition is "spots" machined on the inside of the frame opposite the crank disk and exactly in line with the center line of the engine cylinder. The advantage of these spots is that at any time the alignment of the shaft is questioned it is only necessary to caliper from one of these spots to a point on the crank, then turn the engine over to the other spot and caliper again, and adjust the position of the shaft to bring the distances even. The advantage of this method over running a line through the cylinder is obvious.

To reduce the inertia and the wear on the bottom of the cylinder, the weight of the piston, piston rod and crosshead should be as light as is consistent with ample strength, and in all except the smaller engines the piston rod should be secured to the crosshead by some means other than screwing it into the crosshead and using a locknut, unless provisions are made to prevent the locknut and rod from backing out and breaking the cylinder head. The piston rods should be fitted with some reliable metallic packing, which in many cases may also be used on the valve stems.

Every engine above 100 hp. should be provided with some sort of a turning-over device by means of which it may be easily turned off the center. This feature, of course, appeals more strongly to the operating than to the consulting engineer. In the larger sizes the eccentrics are usually detachable from the valve gear and the valves may be operated by hand and the engine moved by admitting steam through the bypass valve. This is of advantage in starting up and stopping, also in warming up. Sometimes it is necessary to move the crank through a part of the stroke, which can easily be done with a detachable valve gear.

The position of the throttle wheel is another thing that attracts the engineer's attention. Where the detachable valve gear is employed it is usually necessary to have the throttle wheel on the valve-gear side, but not always, as the bypass is used when operating the valve gear by hand. When it is in this position it is necessary for the engineer to stand in line with the flywheel. Suppose the governors stick and the engine is racing badly. This puts the engineer in a dangerous position while stopping the engine. Some builders put the throttle wheel on the opposite side of the cylinder. The valve should turn easily and have spokes outside the wheel rim to make it more convenient to handle. Large Corliss engines have an extension throttle wheel on the valve-gear side of the cylinder, bringing the handwheel well out over the wristplate. Unless a supporting bracket is placed under the stem near the wheel, it will have considerable spring and will bind in the threads and turn hard. This fault alone was the cause of an engine wreck for the reason that although the engineer was at the throttle trying to close it, the flywheel speed increased faster than he could close the throttle and the flywheel exploded; fortunately the engineer was not killed.

Simplicity in the governor and valve gear should have special attention in selecting the engine. It seems to be the aim of some engine builders to see how many unnecessary pieces they can get into this part on the grounds of economy in steam consumption. This causes more friction and more wear, more bearings to look after, more noise in operation, more expense in maintenance and sometimes poor regulation when the parts become worn.

Some of the most economical tests made were on engines with a very simple valve gear and governor.

In selecting an engine the method of lubrication should have close attention. In some of the self-oiling or splash systems of oiling, the oil passages to the shaft bearings, crank and eccentrics are concealed, are complicated, may be easily stopped up if a piece of waste gets into them and a hot bearing may be the result before the engineer is aware that there is anything wrong. The oil in this type of system mixes with molding sand from the inside of the castings, and fine particles of metal are kept well-mixed with the oil which shortens the life of the bearings. The best system of oiling is the gravity feed, in which the oil after being used is filtered and cooled before it goes to the bearings again. Furthermore, the feeds are so located and of such design that the flow of oil may be seen.

Another thing of assistance which is not usually supplied is a full set of working drawings so that the engineer can study every part of the engine without taking it apart and in case any part becomes badly worn or broken he has the drawings for making a new part exactly like the original. This is most often required in rebabbitting the boxes, making piston rings and refacing the crosshead shoes. These drawings should be demanded with every new engine.

Faults can be found in almost every type of engine in use today. Some of these faults affect the economy, others the maintenance and still others the life and safety of the engine. Every engine, however, has its good points as well as bad ones. Both should be taken into consideration in its selection, and the machine should be chosen that seems to have the greatest number of good points and the least number of bad ones. This, of course, requires that the man who selects the engine to be installed must make a study of every feature of those under consideration so that he may know just what he is getting.

Hyattsville, Md. J. C. HAWKINS.

Attention to Brine Strength

I wish to call attention to a condition that might exist in a refrigeration plant using the flooded system in can tanks. When a sample of anhydrous ammonia is drawn off into a tube, the ammonia rapidly boils and evaporates until the tube becomes coated with frost or ice. Evaporation then goes on slowly, as the ice on the tube acts as an insulator and tends to retard the heat passing from the atmosphere to the ammonia. To evaporate the ammonia quickly, the ice must be removed from the tube by the application of water to the frosted part.

Assume that the brine in the ice tanks becomes weak, say as low as 45 deg. salimeter. Such a strength would freeze at about 17 deg. F. Operating the tank at a back pressure of 15 lb., which would give an ammonia temperature of zero deg. inside the pipe in the tank, we would expect, even with good brine agitation, that a coating of ice would be formed on the outside of the pipes containing the liquid ammonia. This coating of course insulates the pipe and tends to decrease the evaporation of the ammonia, just as the ice on the test tube decreased the evaporation of the sample of anhydrous. The result would be that the liquid ammonia in the condenser and receiver would finally be stored in the ice tank and the plant would show indications of being short of ammonia.

As a suggestion, it would be advisable for the engineer, before adding ammonia to the system, to see that the brine strength is up to normal. GAY A. ROBERTSON.

Atlanta, Ga.

Wrong Field Connections

One of the standard instructions covering the starting of a shunt motor is to be certain that the motor field is energized before closing the armature circuit. This applies particularly to cases where the motor has just been installed or where existing connections have been disturbed. In the simplest case, in which the motor field is connected across the line switch so that the field circuit, if intact, takes current as soon as the line switch is closed, the precaution is taken to hold a piece of iron against a pole immediately after closing the line switch and before advancing the starter to the first starting contact. That the indications of even this test may be misleading is indicated by the following.

An operator complained that his 975-r.p.m. constant-speed motor had a speed of 1,600 r.p.m. on the first contact and that the breaker would operate before the

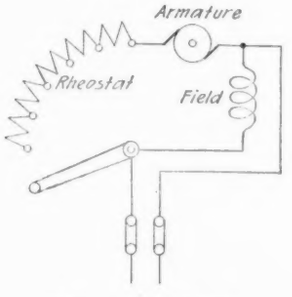


FIG. 1. PROPER CONNECTIONS

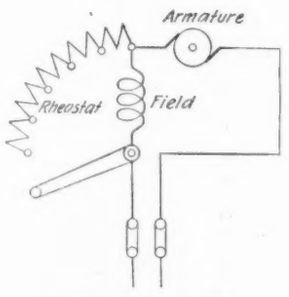


FIG. 2. WRONG CONNECTIONS

starter could be advanced to its full "on" position. Fig. 1 indicates, diagrammatically, the proper connections for such a motor. The field circuit is connected across the line switch, so that the closing of that switch energizes the field and it remains energized throughout the travel of the starter handle, which simply cuts resistance out of the armature circuit. Fig. 2 indicates the connections as they were on the motor that gave the trouble. It should be observed that the field circuit here takes current as soon as the line switch is closed; therefore the test with a piece of magnetic metal would indicate conditions to be normal, but through a circuit that includes the armature. That the armature did not start as soon as the line switch was closed was because the field resistance held the current to a comparatively small value. Upon moving the starter to the first starting contact, however, the armature took more current through the resistance and started, but as this starting resistance was in parallel with the field circuit, the field was weakened. Further advancement of the starter handle cut resistance out of the armature circuit, as it should, but as the starter resistance was decreased, the extent to which the field circuit was short-circuited increased and the field became weaker and weaker; at the same time the armature speed becoming dangerously high and the current heavier, before the starter could be brought to

the full limit of its travel the current became abnormal and operated the circuit-breaker.

It thus becomes evident that although the test with a piece of iron may prove that the motor field is excited, it does not prove that the circuit through which it gets its excitation is a regular circuit; and it does not prove that no danger may be attached to trying to start the motor. In addition to the excitation test, then, where there is any doubt it is well to interrupt the armature circuit by putting paper under the brushes or by disconnecting one of the armature terminals when testing for excitation with the piece of iron. If the motor field remains excited irrespectively of whether the armature circuit is interrupted or not, conditions may be assumed to be normal; but if opening the armature circuit at the armature interrupts the field excitation, it is in order to investigate the connections and correct them.

Schenectady, N. Y. E. C. PARHAM.

Coal and Air Regulation with Stokers

In many cases stokers and the fans for the forced draft are driven by the same engine or turbine. The speed of the engine is controlled by a valve operated by a pressure regulator, the engine speeding up if the steam pressure drops and slowing down when the desired pressure is reached. This method was employed in various plants in which I have worked and is usually advocated by manufacturers. Sometimes, by elaborate tests permanent adjustments are made which, we are told, give "perfect combustion at all times."

This "permanent adjustment" never seemed to me to give satisfactory results. The good points of this automatic control are mainly that the firemen need little brains. But there are objections. If the boilers are driven hard for some time the fires will likely get thin and stay thin. Holes soon appear in the fires, with no way to cover them. Suppose the reverse takes place and the boilers are running with a continual light load. The fires gradually get heavy until it is necessary to pull out the clutches and stop the coal feed. Then the fires burn down all over, and should a heavy demand for steam be suddenly made there is no way to properly meet it. Again, the air from the fan usually is not evenly distributed. This means that one fire burns out faster and soon gets full of holes, while another is choked with coal. Of course gages can be attached to each stoker wind-box, showing the air pressure under each fire, thus allowing for equalizing the air supply; but even this is not satisfactory. Should it be necessary to get a banked fire burned through in a hurry, it is sometimes a tedious job and not always possible in the time available.

In the plants I have in mind, each stoker was equipped with its own variable-speed transmission and driving motor, so that it was possible to operate any stoker at the speed desired.

The results were everything that could be expected. The fan engine driving the stokers was relieved of considerable load, the operator had the fires under control at all times and the proper coal feed was maintained. A banked fire could be quickly burned through, and when cleaning fires they were burned down so that little coke was lost in the ashpit. High CO_2 was at once apparent, the results showing between 12 and 14 per cent. on peak loads. All these advantages naturally increased the capacity of the boilers and also effected a large saving in coal.

The automatic coal and air supply has been changed in plants I have been connected with, in each case leaving the automatic air supply as it was, but allowing the fireman to regulate the speed of the stoker to suit his needs. This gave satisfactory service.

Brooklyn, N. Y. WARREN D. LEWIS.

Centrifugal Boiler-Feed Pumps

In several recent issues (July 27, page 132, Aug. 24, page 275 and Sept. 7, page 347) there have been a number of communications relative to centrifugal pumps for boiler-feed service. Having sold and installed a great many pumps of this type for various purposes, I have formulated the following basic conclusions regarding their use for this service.

Every large plant should use them, because the first cost is less than that of a high-grade outside-packed plunger pump. The degree of reliability is greater than that of reciprocating units, owing in a large measure to entire absence of shock, jar and vibration (which in turn exerts a favorable influence on the maintenance of the unit and the entire system of feed piping), and the fact that they contain fewer working and wearing parts. The cost of operation is considerably less, by reason of uniformly better steam economy and reduced cost of repairs, repacking and attendance. There is freedom from danger of over-pressure on feed lines, which is caused by failure of the pressure governor in the case of the reciprocating machine. This is, of course, in a measure offset by the fact that the speed governor of the driving turbine may also fail. The turbine should, however, be provided with a safety or emergency stop to prevent over-speeding, and some turbine builders are including this feature as a part of their regular construction even on the smaller machines. The pump should be provided with a pressure-relief valve set, say about 25 lb. above the highest feed-line pressure ordinarily carried. Over-speeding or racing will not, however, injure a centrifugal unit to the extent that a reciprocating unit would suffer.

In the smaller plants, having a boiler load of 1,500 to 1,200 hp. and less, the selection of a feed pump presents a problem of greater complexity. The first cost of a centrifugal unit of small size is generally greater, especially if turbine-driven, but motor-driven machines compare more favorably in price. The steam economy of the centrifugal unit is generally better even with the pump-end efficiency as low as 30 per cent. and a turbine economy of 45 to 50 lb. steam per hp.; the total steam consumed for feeding the boilers is less than that required by steam pumps.

As to reliability, I know of a centrifugal of 3-in. size having three stages and a capacity of approximately 150 gal. per min., handling water at an average temperature of 206 deg. F. and pumping against a feed-line pressure of more than 150 lb. After being operated continuously twenty-four hours per day for more than a year, it required no work done on it beyond the renewal of one or two wearing rings and repacking of the stuffing-boxes. A number of pumps are so designed that the stuffing-boxes

never have a higher pressure to take care of than that of the suction line.

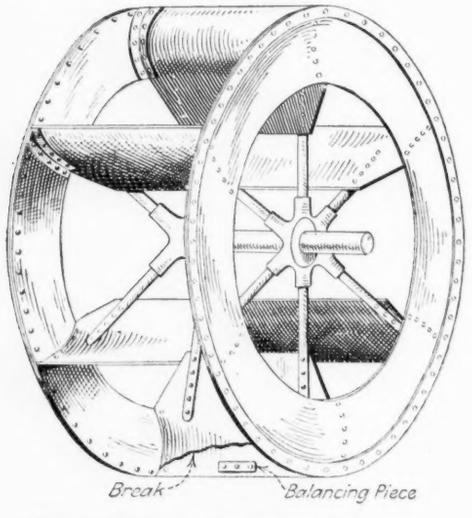
The centrifugal pump should not be too large for the work it is to perform, as is the practice when purchasing a reciprocating unit, and the design of the impellers should receive consideration as well as the material of which they are made and the rigid construction of the machine.

In conclusion, it is my conviction, based upon both experience and observation, that where the conditions warrant, a centrifugal boiler-feed unit will give more satisfaction and a greater degree of reliability and dependability than a pump of the reciprocating type.

Indianapolis, Ind. J. E. KAMPS.

Blades of Two Blowers Break

At the plant of the Atlantic City Gas Co., shortly after the busy season began and within a few days, the blades in two of their three gas blowers broke. In each case the break came without warning and under the usual operating conditions. The importance of these units



TYPE OF BLOWER SHOWING BREAK

and the risk involved by having them out of commission made it very desirable to ascertain the cause of the trouble.

The blowers are used for supplying air blast to the water-gas generators. One is in use at a time, with two in reserve. They are of the steel-plate wheel type blowers and directly connected through a flexible coupling to a small steam turbine. The fan wheels are 30 in. diameter and have six blades carried by two spiders, the arms of which are steel or wrought-iron tee pieces. These arms back up the steel-plate blades, which are 1/8 in. thick, to within about 3 in. of the periphery of the wheel. Beyond the ends of the arms the blades curve backward.

The fans run at about 2,400 r.p.m. during the blast period of the gas-making cycle, which lasts five minutes, and is followed by the steaming period, also five minutes, during which the blower runs idle at about 600 r.p.m., no air being taken from it. This cycle is repeated more than one hundred times a day every day in the week. One blower is usually in operation several weeks at a time and requires little attention other than lubrication.

The breaking of both the blower blades occurred about three hours after the blower had been started on its series of cycles. In one case the break occurred while the blower was speeding up and was followed by a loud roar and violent vibration due to the high speed and unbalanced condition of the wheel, so that nearly all the bolts were loosened before the fan could be stopped, and the babbit in the outer bearing was badly cracked.

An examination of the fan wheel in each case showed that a large piece had been broken out from the outer end of one of the blades beyond the extremities of the spider arms. What appeared to be an old crack extended several inches in a direction roughly parallel to the shaft. From either end of this crack the final break had extended diagonally to the outer corners of the blade. It was further noted that each section broken out had a piece of bar iron about 3x1x1/4 in. riveted to it close to and parallel with its outer edge, presumably for the purpose of balancing the wheel.

The wheels were returned to the manufacturer for repair, and his opinion was that the break was caused by inertia. He advised the substitution of straight blades for the curved ones, better to resist the strains. He did not state why the original design failed under the service for which it was made, nor did he comment on the influence, if any, of the counter-balances placed as they were, although his attention was directed to them. We note, however, that the rebuilt wheels have the counter-balances on the side rims and not on the blades, as before. The rebuilt wheels have not yet been used, so we cannot predict their behavior.

Possibly some of the readers of *Power* have had similar experiences and a discussion of the points involved would be valuable. WILLIAM A. DUNKLEY.

Atlantic City, N. J.

Inspiring the Firemen

An engineer in a plant I have in mind fires up one of the furnaces whenever he feels like taking exercise. When no other duty calls his attention he fires and experiments, tries new grates with various-sized openings, adds to or decreases from the height of the bridge-wall and tries various damper openings for different depths of fuel. And he always tinkers with that boiler—no other. He is careful not to make a nuisance of himself among the firemen, and he has succeeded in making himself so interesting to these shovelers that they interrogate him and copy his methods. The result is that the coal bill up to June was much less this year than last. He told me that the reduction so far was 22 per cent. and that he expects to improve on this record in the coming year.

This engineer never lets a day pass without firing an hour or so. He has read nearly everything on the subject of combustion and applies his knowledge to the practical everyday conditions of the boiler room. He has learned to fire by firing—and firing—and firing.

Before engineers can reasonably expect much improvement in the work of their firemen, they should not forget that someone must furnish the inspiration. But to acquire the ability to inspire the firemen, the engineer must first prove to them that he is a better fireman than they are. LUKE MARIER.

Fall River, Mass.

Inquiries of General Interest

Source of Expansive Force in Gas Engine—Is the expansive force of gas in the cylinder of a gas engine derived from greater space required by the change in form of the gas, or is it due to heat of combustion of the mixture?

C. B. H.

The expansive force and work performed are due to the increase of volume attending the rise in temperature when the mixture burns.

Rim Speed of Pulley—What is the rim speed in feet per minute of a pulley 30⅜ in. diameter running at 657 r.p.m.?

L. E. D.

The rim speed in feet per minute would be the circumference in feet multiplied by the revolutions per minute, or

$$\frac{30\frac{3}{8} \times 3.1416}{12} \times 657 = 5{,}224.57 \text{ ft. per min.}$$

Equal Pipe Areas—How many 1¼-in. pipes would be required to have a united cross-sectional area equal to one 8-in. pipe?

R. C. C.

The areas of circles are to each other as the squares of their diameters, and therefore the number of 1¼-in. pipes required to equal one 8-in. pipe would be

$$\frac{8 \times 8}{1.25 \times 1.25} = 40.96, \text{ or practically forty-one } 1\tfrac{1}{4}\text{-in. pipes.}$$

Cost of Developing Power—What would be the cost of developing 275.75 hp. for 9½ hr. if the boiler uses 4.3 lb. of coal per horsepower per hour and the price of coal is $4.65 per ton of 2,240 lb.?

T. D.

As there would be 275.75 × 9.5 = 2,619.625 hp.-hr., the coal consumption would be 2,619.625 × 4.3 = 11,264.4 lb., and as the price of coal would be $4.65 ÷ 2,240 = $0.002076 per lb. the cost of coal would be 11,264.4 × $0.002076 = $23.38.

Size of Supply for Exhaust-Steam Heating—What should be the size of an exhaust-steam main for supplying 2,000 sq.ft. of heating surface?

G. D. W.

To cover all conditions it would be well to make the supply main, at the first point of distribution, of a size appropriate for low-pressure gravity-return steam heating, which should have a diameter, in inches, equal to about the square root of $^1/_{100}$ of the heating surface; that is, $\sqrt{\frac{2{,}000}{100}}$ or about 4½ in. dia. From the engine to the first point of distribution the size of the supply pipe should be greater, according to the length and the number of elbows.

Putting Out Fires in Coal Piles—What causes spontaneous combustion in coal piles, and how can fires in coal be extinguished?

J. W. R.

The spontaneous combustion is due primarily to the rapid absorption of oxygen by the coal and to the oxidation of iron pyrites (fool's gold) occurring in the coal. The conditions favorable to this process are a supply of air sufficient to furnish the necessary oxygen, though insufficient to carry off the heat generated; finely divided condition of coal presenting a relatively large surface for absorption of oxygen in proportion to masses of pieces oxidized; and large percentage of volatile matter in the coal and high external temperature.

Fire in the interior of a pile can be extinguished by forcing water or steam into iron pipes that are provided with drive-well points and driven down into the coal pile to places where fires exist.

Scale for Mercury Vacuum Gage—How can a U-tube mercury gage be graduated for use in checking against the readings of an ordinary spring vacuum gage on a vacuum heating system?

J. M.

The line of common level of mercury in the legs of the U-tube, when the surface of the mercury in each is acted on by the atmospheric pressure, will be the location of zero inches of vacuum, and the inches of vacuum will be the sum of the differences of mercury level above or below such zero. But the zero level will fluctuate with changes in the density of the mercury, owing to variations in temperature, and in place of having a movable scale, it is better to have a fixed scale with its zero placed at the approximate zero level, with graduations laid off in inches and fractions of inches above and below the zero division. Then, as the inches of vacuum to be read is the difference of level of mercury in the legs of the U-tube, the reading is found by simply adding together the readings taken above and below the arbitrary, or fixed, zero.

Water Under Pressure Discharged Through Aperture—How many gallons of water per hour would be discharged through a clean ½-in. dia. hole in the shell of a steel tank under 40 lb. gage pressure?

H. V. M.

The flow in cubic feet per second is given by the formula

$$Q = c\sqrt{2gh} \times a,$$

in which

Q = Cubic feet discharged per second;
c = a coefficient which for the conditions may be assumed as equal to 0.62;
2 g = 64.4;
h = The head or height in feet that would be equal to the pressure;
a = The area of the aperture in square feet.

As 40 lb. pressure per square inch would be equivalent to a head of 40x2.3 = 92 ft., then $\sqrt{2gh} = 76.9$, and as the area of the aperture ½ in. dia. would be $\frac{0.19635}{144}$ sq.ft. and a gallon is 231 cu.in., the discharge would be

$$0.62 \times \left(76.9 \times \frac{0.19635}{144}\right) \times \frac{1{,}728}{231} \times 60 \times 60 = 1{,}750.7 \text{ gal. per hr.}$$

Steam Consumption from Indicator Diagram—What is the formula for estimating the steam used by an engine as determined from an indicator diagram?

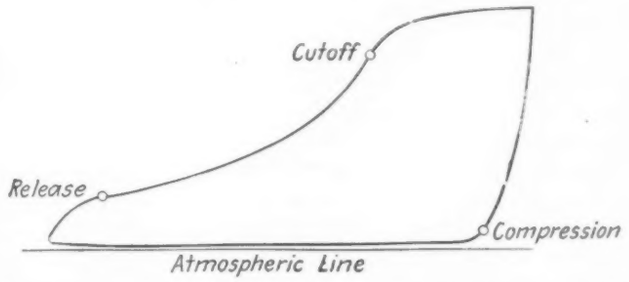
W. R. C.

The steam accounted for by an indicator diagram, expressed in pounds per indicated horsepower per hour, is found by using the formula

$$\frac{13{,}750}{\text{m.e.p.}} [(C + E)\,Wc - (H + E)\,Wh]$$

in which

m.e.p. = Mean effective pressure;
C = Proportion of stroke completed at a point selected on the expansion line which should be taken near the point of actual cutoff or near the point of release;
E = Proportion of clearance;
H = Proportion of stroke uncompleted at a point selected on the compression line closely following the point where the exhaust valve is completely closed;
Wc = Weight of 1 cu.ft. of steam at the pressure of the point selected on the expansion line corresponding with C;
Wh = Weight of 1 cu.ft. of steam at the pressure of the point selected on the compression line corresponding with H.



POINTS FOR CONSIDERATION OF STEAM CONSUMPTION

The points of cutoff, release and compression referred to are indicated in the figure.

Sudden Cooling of Boilers in Cases of Low Water

By R. N. Blackburn*

SYNOPSIS—A defense of the statement that the sudden cooling of overheated boiler plates is the safest practice in cases of low water.

In a bulletin recently published by the Steam Boiler Inspection Department of Saskatchewan, Canada, operators of steam boilers are advised in case of low water to "shut the draft damper, open the fire-door and start the injector or feed pump." As this advice is opposed to the generally accepted rule of conduct in such cases, a statement of the principles involved and the effects produced follows:

The bulletin was published for the use of owners and operators of steam traction engines, and in deciding to give the foregoing advice, special consideration was given to the conditions under which this class of steam boiler is usually operated, and there were special reasons for the advice given. It is believed, however, that the advice to inject feed water into the boiler as soon as the water is discovered to be dangerously low is in most cases the safest rule to follow for all types of steel boilers. The idea that admitting feed water to a boiler in which the water is dangerously low may be disastrous has at least the merit of antiquity.

Dangers Attending Low Water

A boiler in which the water is too low is usually in a dangerous condition and is likely to explode at any time, whether the feed water is turned on or not. It is of course well known that if cold water be thrown onto a hot brittle substance, such as a heated glass vessel, or a red-hot cast-iron plate, the result will frequently be to fracture it; and if a boiler in which the water is low should happen to explode at the time the feed water is being turned on or at any time after it was alleged to have been turned on, the disaster has, plausibly, usually been attributed to the action of the fireman in admitting feed water. It is quite possible that the turning on of the feed water may not have been contributory to the accident; but as the fireman is usually unable to participate in the subsequent investigation, if there is one, the blame may frequently be laid on him without contradiction.

A steel boiler plate is not likely to fracture even if cooled rapidly. The standard specification adopted by the Association of American Steel Manufacturers requires that this material shall be capable of being heated to a cherry-red, quenched in water and afterward bent double without fracture. It must be admitted that this is a much more severe test than merely cooling the heated boiler plates by admitting feed water.

When the water in a steam boiler is discovered to be dangerously low, the usual practice is to draw the fire or smother it. If the furnace crown has already begun to bulge, it is doubtful whether the cooling will be rapid enough to prevent its collapse, with possible serious injury to the fireman. This is especially the case when the furnace crown is flat or nearly so, as in a locomotive-type boiler, or where a mass of heated brickwork continues to radiate heat to the boiler. Many cases are on record of men having been killed while attempting to draw the fires in cases of low water, and it seems desirable to adopt some safer and, if possible, more rapid way of cooling the overheated plates and restoring normal conditions.

When a crown-sheet has become overheated on account of low water and is presumably on the point of collapsing, the most obvious thing to do is to restore its strength by cooling it as quickly as possible. Measures which could be taken with this object in view, in addition to drawing the fires, might include turning on the feed water and easing the safety valves so as to induce foaming, which would cause the rising froth to cover and cool the overheated plates.

Most boilers are, or ought to be, provided with a fusible plug, and in most cases, when this plug has not melted, the furnace crowns will not yet have become unduly overheated and there can, therefore, be no risk in admitting water, even if the level in the boiler is below the visible bottom end of the water glass. It is possible that, owing to neglect in keeping the fusible plug free from scale, it may be inoperative and the crown-sheet have become considerably overheated before the water is discovered to be too low. It is sometimes asserted that the fresh feed making contact with the red-hot plates would generate an explosive gas. The amount of gas that could be thus liberated would be negligible.

Stresses Caused by Cold Water

It is sometimes claimed that the sudden contraction of the overheated plate by the admission of feed water will cause it to tear away at the nearest riveted joint. The local overheating brought about by low water has already caused the heated portions to expand, inducing severe stresses in adjacent parts. When the overheated plates are cooled by the incoming water they contract to practically their original dimensions, relieving the stresses produced by the previous expansion and so restore normal conditions. The contraction, therefore, instead of increasing the stress at the riveted seams, may reduce it. Cooling the plates by raising the water level will occur in the reverse order to the expansion caused by overheating. The boiler is enabled to take up the distortions in such order as best to relieve the stresses in the plates so that stresses produced by cooling will not exceed those caused by the previous overheating, which, by hypothesis, have not been sufficiently severe to cause rupture. The importance of this may be illustrated: If a flat plate is securely riveted all around its edges to a heavy and rigid frame and the plate is heated while the frame remains cool, the expansion of the plate will produce stresses at the seams around its edges, which stresses may be assumed to be insufficient to induce fracture, as the expansion of the plate is in great measure taken up by the plate buckling. The heating is now continued until the frame becomes hot and by its expansion relieves most of the stress in the plate.

*Chief inspector steam boilers, Saskatchewan, Canada.

If, now, we suppose that first the frame and then the plate be cooled, the first effect will be to contract the frame and rebuckle the plate, reproducing the conditions of stress obtaining when the plate was originally heated and which we have assumed not to be sufficiently severe to cause fracture. If we suppose the plate to be cooled first, the frame remaining hot, the stress set up by the contracting of the plate would be much more severe on account of the rigidity of the frame and might cause the plate to tear away from the frame. In this case, if the frame had been heated first, rupture would have occurred during the initial heating and the stresses produced by subsequent cooling would not enter into the question.

It must also be remembered that the several parts of a boiler are not held so rigidly as has been assumed. Provision must be made for expansion and contraction due to changes of temperature, and there is usually considerable "breathing action" when the boiler is in operation, which would be sufficient to allow for any contraction caused by cooling the overheated plates, without causing undue stress.

Explosion from Over-Pressure

Another objection frequently advanced is that the water coming in contact with the red-hot plates would generate so much steam as to explode the boiler from over-pressure. This objection may be conveniently considered by assuming an actual case. A slight fall in the water level is usually more dangerous in internally fired boilers than in the externally fired horizontal return-tubular type. This is especially the case in boilers that have the top of the firebox or combustion chamber only a little below the working water line, as in a locomotive-type boiler. In this type a slight fall in the water level will expose all or nearly all the crown-sheet, and owing to the large area of the plate which becomes overheated, it may be assumed that the amount of steam generated by admitting fresh water will be proportionately greater than in most other types. We will, therefore, take as an illustration a well-known locomotive-type boiler—for example, a J. I. Case traction engine. The boiler referred to is 33⅜ in. diameter in the barrel, the firebox 44x30⅝ in. inside, and the boiler is allowed in this province a working pressure of 150 lb. The crown-sheet is arched as required by the Saskatchewan Regulations; and if we assume that the water level has been allowed to fall 2 in. below the top of the crown-sheet, the exposed portion of this sheet above the water line would measure 44x24 in. Assume that the whole of the crown-sheet above the water line has been heated to a bright-orange heat, say 2,100 deg. F., and that feed is then admitted. This is an extreme case, as in practice the crown-sheet would have probably collapsed before this condition had been reached. If, now, the feed water be turned on, it will require 250 lb. of feed to raise the water level to the top of the crown-sheet, and assuming the temperature of the incoming feed to be 110 deg. F., 64,000 B.t.u. would be required to heat the incoming feed to the temperature of the water in the boiler. As the crown-sheet in cooling would give out only 19,950 B.t.u., it is evident that admitting feed will reduce the temperature and consequently the steam pressure. If the feed is taken in at 212 deg. F., it would still require 38,500 B.t.u. to raise it to the temperature of the water in the boiler. This is nearly double the amount of heat given up by the crown-sheet, so the steam pressure would be reduced rather than increased. Even if the boiler had a flat crown-sheet so that the area of the heated surface was equal to the full area of the fire-box—namely, 44x30⅝ in.—the total quantity of heat given up by the crown-sheet in being cooled to the temperature of the water would not exceed 25,300 B.t.u., which is less than the amount required to raise the incoming water to the temperature of the water in the boiler.

Danger Slight under Worst Conditions

It may be objected here that, owing to imperfect circulation, the incoming feed, instead of being thoroughly mixed with the water already in the boiler, might settle in the lower part of the boiler and not be immediately raised in temperature. This would displace a corresponding amount of the heated water, which would come in contact with the heated crown-sheet and, being already at the boiling temperature, would generate a considerable amount of steam. Take an extreme case and assume that the incoming feed has no cooling effect on the boiler and that the water in the boiler, which is already heated to 365 deg. F., is converted into steam as rapidly as it can be displaced by the fresh feed forced into the boiler.

The boiler is provided with a ¾-in. injector that has a delivery capacity of from 170 to 355 gal. per hr. As we have assumed the feed water to be evaporated into steam as quickly as it is forced into the boiler, it is clear that under these conditions steam would be generated for a time at the rate of from 1,416 to 2,958 lb. per hr. The boiler is provided with a 2-in. safety valve. We may safely assume that any 2-in. safety valve of good make will easily discharge this amount of steam without material rise in pressure. One well-known maker of safety valves, for example, guarantees his 2-in. valves to have a relieving capacity of 4,410 lb. of steam per hr. at 160 lb. pressure. (At 150 lb. pressure the relieving capacity would be about 6 per cent. less.) Since the relieving capacity of a safety valve increases with the pressure, and remembering that this rapid generation of steam can continue only for a very short time, that is, until the crown-sheet has been cooled, we may conclude that even if the feed pump, with which the engine is provided in accordance with the provincial requirements, were also started in addition to the injector, a fairly efficient safety valve would, without a serious rise in pressure, release the steam generated.

Safety Valve Capacity Sufficient

Taking a more extreme case and assuming that the heated crown-sheet could be instantly flooded, steam would not be generated instantaneously. When a red-hot piece of boiler plate is plunged under water, it takes an appreciable time to cool, and if the water has access to one side of the plate only, as in the case we are considering, the time required is increased. The total weight of steam generated in cooling the crown-sheet will not under the most extreme conditions exceed about thirty pounds, and as it would probably require thirty seconds to cool the plate, a fairly efficient safety valve would discharge the steam without a dangerous rise in pressure.

In considering the best course to adopt in cases of low water, it has been assumed that the boiler is provided with a proper safety valve in good order. Where a so-called engineer permits the safety valve to become inoperative, allows the fusible plug to become so scaled over as to

be useless or plugs it with an iron bolt as is sometimes done, and then neglects to keep sufficient water in the boiler, this method of dealing with low water is likely to be attended with risk.

In 1867, on account of conflicting opinions as to the cause of some boiler accidents, experiments were carried out in England to ascertain the effect of injecting cold water into domestic heating boilers that had been allowed to become overheated by shortness of water. It was found that no explosion followed.

In 1889 the Manchester Steam Users' Association carried out further experiments with a full-sized high-pressure Lancashire boiler. This boiler was 27 ft. 9 in. long by 7 ft. diameter and was provided with two internal furnaces, each 3 ft. diameter. The boiler was of the usual type of that date, with plain circular furnaces, lap-jointed and single-riveted, without stiffening rings. The feed pump used was capable of supplying four times the amount of water usually evaporated. The object of employing so large a pump was to reproduce the conditions where one of a battery of boilers became short of water and the feed supply for the whole range might be turned into the affected boiler. A series of thirteen tests was made, with the object of ascertaining the result of turning on the feed in cases of low water. After raising the steam pressure, the water was lowered to the level of the furnace crowns by means of the blowoff valve; the valve was then shut and the fires kept briskly burning, thus causing the water level to be further lowered by evaporation. At the expiration of a specified time the feed was turned full capacity on to the furnace crowns. In every test the steam pressure was reduced, although it was proved from the condition of fusible disks previously placed on the crowns, that the latter in some cases had been red-hot. After seven tests had been made, the maximum waiting time being 15 min., two further tests were made, in which the waiting time was 23½ min. It was then found that the furnaces had actually begun to bulge and that the boiler leaked freely at the joints; but nothing serious resulted from turning in the feed water. The furnace that suffered the least was the one that had been deluged with cold water.

In the first test made, the feed water was not injected soon enough and one of the furnaces collapsed by overheating just at the moment the order was given to turn on the feed, the steam and water rushing out with great violence through an opening 3 sq.ft. in area.

It was found also that when the water level was lowered by means of the blowoff valve to a point considerably below the furnace crowns, so as to expose rapidly a large surface to the fires, a sudden rise in steam pressure occurred when the feed water was first admitted, followed by an almost immediate fall to less than the original pressure. The furnaces were not rent by the sudden contraction, no explosion followed, nor was an uncontrollable rise of pressure suddenly generated.

Developments in Modern Water Turbine Practice*

By Dr. H. Zoelly†

SYNOPSIS—Modern design of water turbines for low-, medium- and high-head developments has solved the problem connected with installations of plants for large output, high speeds and with accurate governing devices. The paper discusses the methods by which such results have been obtained and briefly describes typical installations in various parts of the world.

The first large low-head water-power plants in Switzerland were laid out in 1894, at Rheinfelden on the Rhine, just above Basel, where twenty vertical turbines of 900 hp. each were installed, and at Chevres, near Geneva on the Rhone, with fifteen turbines of 1,200 hp. each. Quadruple turbines were installed at Rheinfelden and double turbines at Chevres, each runner containing three rows of moving blades. The runners were of the conical type controlled by hydraulically operated gates. The revolving weight of the units in both plants is supported by a thrust bearing running on a film of oil under pressure.

At this early date the vertical-shaft generator was exceptional. The first large plant was built at Wangen in 1904 and contained seven vertical quadruple turbines of 1,500 hp. each. Another large development, Fig. 1, was started at Laufenburg, Switzerland, about 1907 and was equipped with horizontal quadruple turbines with inspection chambers for the bearings, accessible from underneath. The controlling mechanism of these turbines was under water, in the open chamber directly connected to the headrace. The elevation of the shaft above the tailrace is determined by the maximum draft head of the water turbines. The generator-room floor is overflowed during floods, so that precautions had to be taken in the construction of the plant to avoid injury to the apparatus. In later installations, however, this trouble is avoided by raising the level of the turbine to nearly the level of the headrace or even above the latter. In these plants

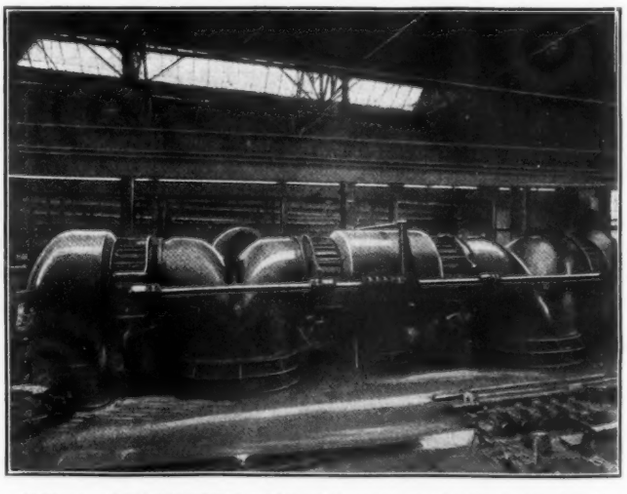


FIG. 1. QUADRUPLE HORIZONTAL TURBINE AT THE PLANT AT LAUFENBURG, SWITZERLAND

the turbines start as soon as the gate is opened, the highest point of the runner being no higher than the headrace level. In a German plant installed eight or ten years ago a sextuple horizontal turbine is above the headrace level, so that a starting device is necessary to set the turbine in motion. By

*From a paper presented at the International Engineering Congress, San Francisco, September, 1915.

†Engineer, Escher, Wyss & Co., Zurich, Switzerland.

opening a small valve, water from the headrace is discharged into the draft tube of the turbine; the water acts as an ejector, quickly producing the necessary back pressure in the turbine chamber to permit the admission of sufficient water to drive the turbine. The time necessary for starting is only two or three minutes. The high elevation of the shaft permitted the direct coupling of the generators to the turbine and avoided all transmission-gear losses which would have

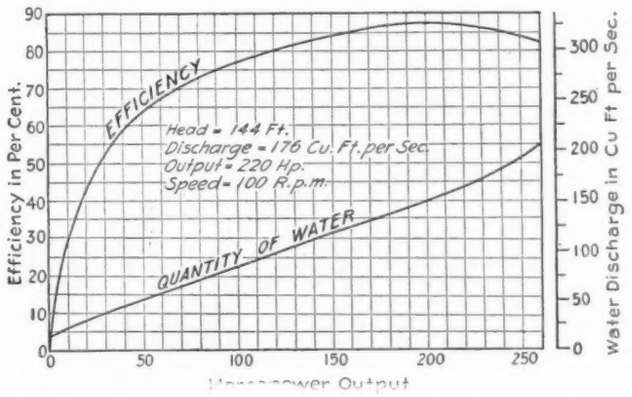


FIG. 2. PERFORMANCE CURVES OF LOW-HEAD TURBINE

been necessary if the water turbine had been installed according to the usual practice.

Twin and quadruple turbines are complicated machines requiring a large number of spare parts. This results in rather high operating costs. These turbines have the disadvantage of exposing the whole controlling mechanism to severe wear as it must work under water. The ideal turbine is one of simple design that will permit the controlling mechanism to be located out of the water. The first improvement was obtained about 1910 and was applied in the large plants at Keokuk on the Mississippi and at Cedar Rapids on the St. Lawrence. The turbines in these plants are of the single-runner type, in which the specific velocity has been increased sufficiently to permit the direct coupling of turbines with generators.

A large plant at Eglisau, the installation of which is now being discussed, will probably contain seven 6,000-hp. units of the vertical single-runner type. The installation of vertical turbines permits the generator to be placed at such an elevation above the tailrace level that the flooding of the generator room, even at high water, can be avoided.

While the main difficulty in designing low-head turbines is to arrange them for sufficient capacity, especially with the smallest heads (that is, during floods), this condition ought to be fulfilled without interfering with the best efficiency during the low-water period, corresponding to the high head with which the runner has to work. Fig. 2 shows the performance of a standard modern low-head turbine as to efficiency and discharge with respect to the output.

The controlling mechanism of the guide vanes of the spiral Francis turbines, the waterwheels most used for medium pressure, originally was always in the water. All the levers, links and pins were constantly exposed to the water stream and its impurities. An installation put down with controlling mechanism in the water is the Brillanne Power House in southern France, with five horizontal twin turbines in spiral casings of 3,500 hp., started in 1906. The advantage of the external regulating device is so great that these turbines were changed in 1911 to this new design. Twin turbines of 7,000 hp., with vertical shafts and spiral casings and external regulation, are running at 128 r.p.m. under a head of 60 ft. in the Kellner Partington installation in Norway.

In installations of high power, large water capacities and short penstocks, the turbines are arranged not in spiral, but in cylindrical casings. The latter design greatly reduces the floor space necessary as compared with that required for spiral turbines. The cylindrical-casing turbines are designed with the controlling mechanism in the water, and although it is not absolutely impossible to arrange the mechanism outside of the casing, it is not generally done because of the increased cost. The power plant of the Western Canada Power Co. at Stave Lake has four horizontal twin turbines in cylindrical casings developing 13,500 hp. each under a head of 115 ft. One of these turbines is shown in Fig. 3 with the upper cover removed. The horizontal shaft at the upper part of the illustration is a part of the controlling mechanism.

The increasing heads for which Francis turbines have been designed require another kind of controlling mechanism, and medium-pressure turbines had first to be provided with external regulating devices. Each guide vane is made in one piece with its pin, the latter extending outside the casing through stuffing-boxes. All the levers are fixed on the pins and controlled by the servomotor by a device consisting of rings and rods placed outside the casing. With this arrangement the working parts are easily accessible, they can be well lubricated and any severe wear and tear is obviated. An example of the externally controlled turbine is shown in Fig. 4, which is one of the six horizontal twin spiral turbines of 17,500 hp. each working under a head of 400 ft. at the municipal plant of the City of Tokyo, Japan.

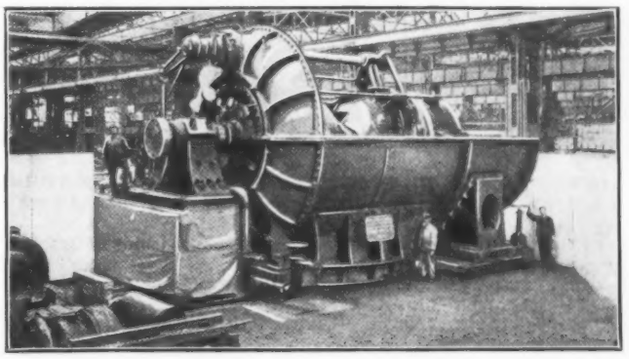


FIG. 3. TWIN HORIZONTAL FRANCIS TURBINES; STAVE LAKE POWER PLANT, VANCOUVER, B. C.

About the year 1910 the manufacturers, in order to decrease production costs, began to use an overhung runner-wheel mounted on the end of the generator shaft. As the controlling

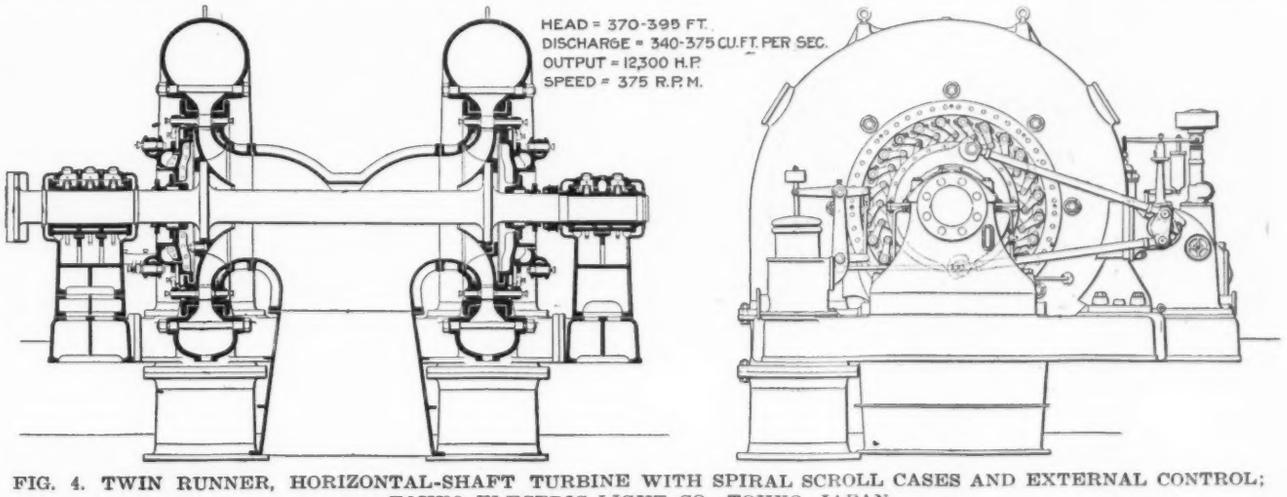


FIG. 4. TWIN RUNNER, HORIZONTAL-SHAFT TURBINE WITH SPIRAL SCROLL CASES AND EXTERNAL CONTROL; TOKYO ELECTRIC LIGHT CO., TOKYO, JAPAN

mechanism can be located on the cover of the turbine and the draft tube bend designed to suit the shape of high-head runners, it is possible to increase materially the specific speed of the turbines without interfering with the accessibility of the external controlling gear. The tube bend has no internal obstruction, such as shaft or stuffing-box, which is an important point in regard to good efficiencies with high-speed runners.

The plant at Tuxpango, Mexico, contains two generator turbines of 9,000 hp. each, as shown in Fig. 5, operating under a head of 558 ft. When twin turbines are used it is possible to disconnect them into separate halves, each runner being connected at one end to the generator shaft, thus providing for a generating set with only two bearings. By dismantling the draft-tube bends, it is possible to remove the runners and change them. Two sets of runners to suit varying heads may thus be used, insuring the best efficiency at all times.

The constant increase in size of generating units for turbines of medium heads, say from 20,000 hp. upward, has resulted in the installing of vertical instead of horizontal turbines, because the large spiral casings are much easier to install in a horizontal than in a vertical position. In fact, the horizontal position of the casing is the only practicable one for units from 40,000 to 50,000 hp. This arrangement is shown in Fig. 6, which represents a 15,000-hp. single-runner spiral turbine working under a head of 165 ft. in the power house of the Barcelona Traction, Light and Power Co., in Spain. The spiral casing is made of wrought iron of a circular cross-section. The whole turbine is fixed on a heavy cast-steel ring supporting directly the yoke and the thrust bearing. The scroll casing of sheet iron is also fixed to this ring, but bears only its own weight and not any part of the weight of the turbine. The thrust bearing, which is of course an important part of the turbine, can be installed on the top of or below the generator, supported by the turbine itself, by yokes or by any similar part. Plans are already made for similar turbines of from 50,000 to 80,000 hp. in one runner with spiral casing of wrought iron.

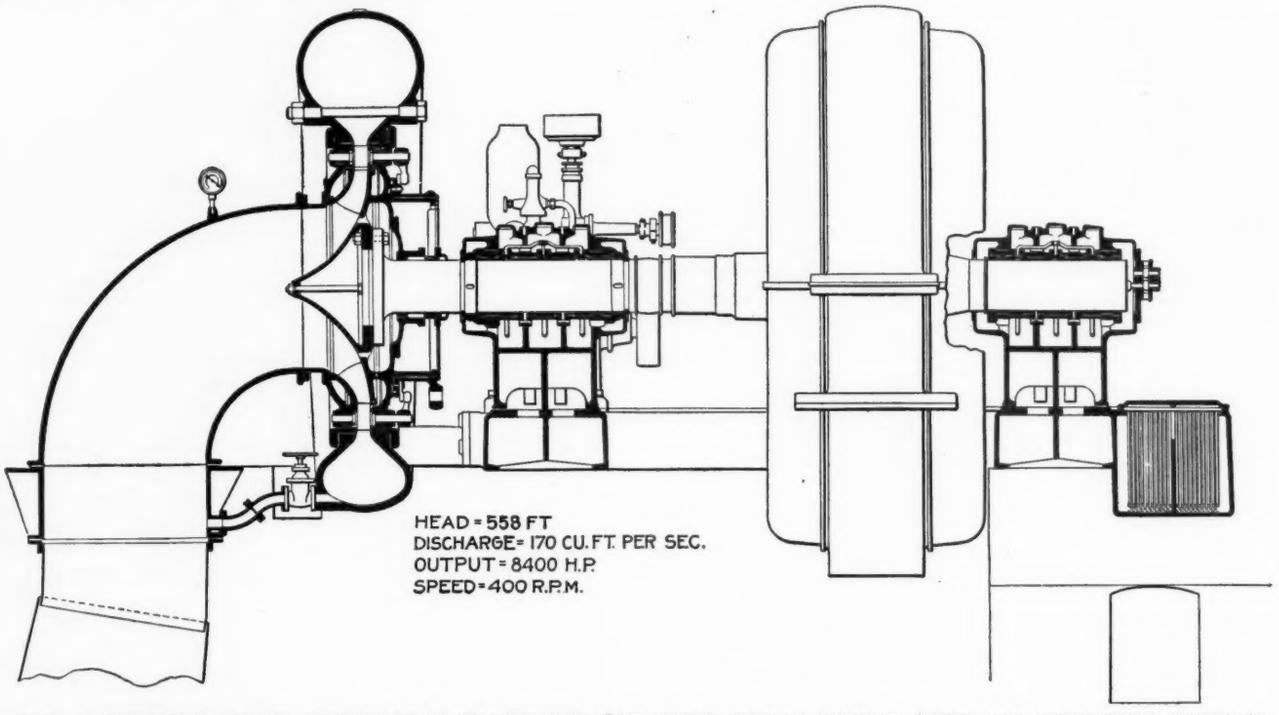


FIG. 5. SINGLE-RUNNER, HORIZONTAL-SHAFT TURBINE WITH SPIRAL SCROLL CASE AND EXTERNAL CONTROL HEAD 558 FT.; TUXPANGO, MEXICO

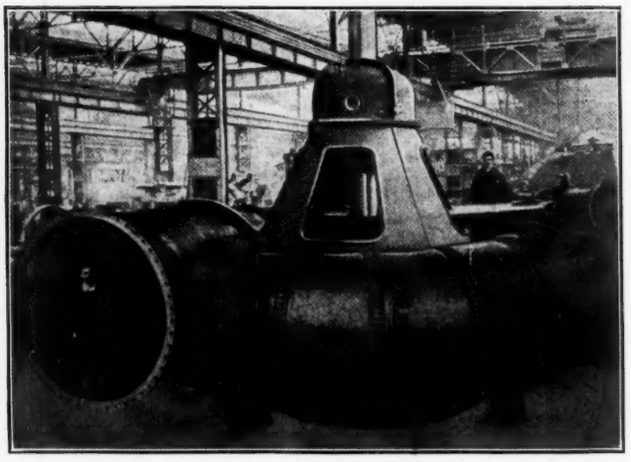


FIG. 6. SINGLE VERTICAL TURBINE IN WROUGHT-IRON SCROLL CASING; SEROS PLANT, SPAIN

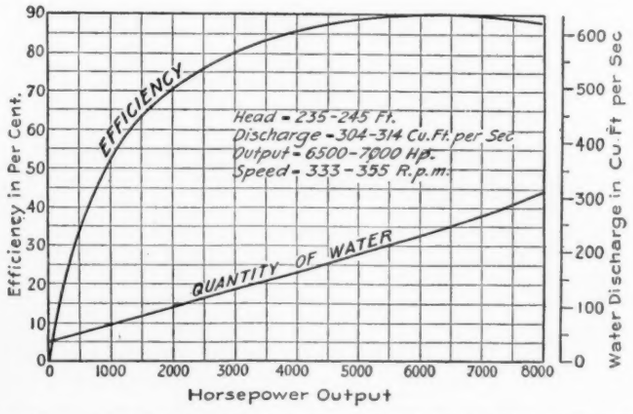


FIG. 7. MEDIUM-HEAD TURBINE PERFORMANCE CURVES

The runner is the most important part of designs for such enormous outputs. Excessive erosion by sand or corrosion caused by impurities in the water must also be carefully guarded against. It is now standard practice to provide medium-head runners with vanes of wrought iron, cast in steel rings and disks. The runner is made of cast steel in one piece for heads of from 165 to 200 ft. Another difficulty to be considered is the transportation by rail, necessitating large runners being made in several parts. These requirements have been satisfactorily met, however, as the runners of the Keokuk plant were made in different pieces without any trouble resulting. At the present stage of water-turbine design it is possible to deal with the enormous out-

puts mentioned with single-runner units. Fig. 7 shows the efficiency attained by a turbine of first-class design.

For about twenty years the Girard turbine has not been considered the best for high heads, as the guide apparatus and the runner of the Pelton wheel is, in principle, better adapted. Since the beginning of this century the latter type has practically held the field in high-head water-power developments.

The first important high-pressure water-power plants were installed in Switzerland. They were equipped with relief valves designed for reducing the over-pressure arising in the penstocks after the sudden throwing off of load. One of the earliest plants working under the high head of 1,300 ft. was installed at Brusio, Switzerland, in 1905. The twelve horizontal turbines are single-jet and develop 3,500 hp. each when running at 375 r.p.m. The relief valves are designed as slide valves and are controlled by a dashpot.

The vertical-shaft arrangement of the Pelton wheel has marked advantages, especially where several jets are necessary to develop the output available. This new design was adopted for the water-power plant of Rio de Janeiro, which contains six 9,500-hp. and two 19,000-hp. turbines of the vertical-shaft single-runner multiple-jet design, all of which have release valves. The thrust bearing has been installed on the intermediate floor below the generator.

Relief valves are not satisfactory for heads of, say, 1,600 ft. and upward as the high pressure practically entails the use of double the number of parts exposed to severe wear. The turbine with a single jet, for instance, has two outlets exposed to the full velocity of the water. A single outlet for the water, acting as nozzle and relief valve at the same time, has been tried in order to avoid any dangerous pressure in the pipe line. This device diverts the jet from the runner at the same instant the load is taken off the turbine, thus preventing racing, and then slowly closes the nozzle, reducing the flow of the water to zero. The outlet, of course, must be closed slowly to avoid any dangerous pressure increase. The first hydraulic engineer in Europe to apply this idea was Zodel, the designer of the deflecting nozzle. This has been adopted at the plant on the River Flamisell, in Spain, which contains four 8,200-hp. turbines running under a head of 2,700 ft. One of these turbines is shown in Fig. 8.

The governor acts through a servomotor on the needle of the nozzle, and by the means of another servomotor controlled by a dashpot acts likewise to control the deflecting nozzle itself. The dashpot acts to return the nozzle to its original position. Let us consider the position of full load. When suddenly taking off the full load the dashpot of the second servomotor remains locked, so that the deflecting motion is effected in the shortest possible time, the jet being deflected without being changed. The position of the gear that operates the needle is immediately changed in accordance with the new load of the turbine. The second servomotor is in the meanwhile acting slowly, owing to the action of the dashpot, and gradually turns the whole deflecting mechanism into its original position. The time required for this combined movement varies in each installation according to the special conditions of the penstocks, but is arranged so that the resulting pressure is hardly perceptible. When taking off the load slowly, the needle alone will be moved. Since the dashpot does not influence the second servomotor, the latter likewise remains inert when the load is added, as the needle alone is moved. The developing of this idea has led to the introduction of a new element, the deflector, which is placed between the outlet of the nozzle and the wheel itself and deflects the jet from the wheel as shown as the load is suddenly removed. The nozzle is fixed, and the deflector is thus suitable for turbines with more than one nozzle.

The modern tendency is to increase the specific speed of Pelton wheels as much as possible, especially for heads of from 1,000 to 1,600 ft., with the result that a less expensive electrical equipment is needed. The ratio of the diameter of the jet to the diameter of the wheel is increased as much as good efficiency will permit, avoiding wherever possible turbines with more than one nozzle. The 8,250-hp. turbine of the Borgne installation running under 1,100 ft. head and 300 r.p.m. with only one nozzle, has the largest jet (about 8¼ in.) that has been made up to the present. The increase in the diameter in the jets naturally calls for an increase in the dimensions of the buckets. These must be fixed to the disks, taking into consideration the strong centrifugal force and severe shocks, caused by the water jets to which they are exposed. It is also important to arrange the buckets in a complete rigid ring around the disk. These difficulties are now overcome, and Pelton wheel units of the enormous output of 25,000 hp. are already under consideration. A glance at Fig. 9 will show the range of efficiency of good Pelton wheels.

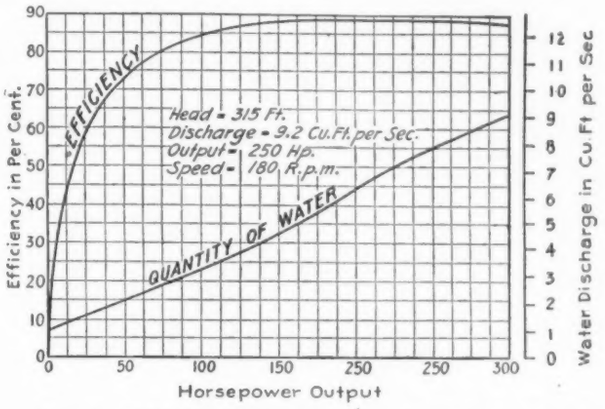


FIG. 9. PERFORMANCE CURVES OF A PELTON OR IMPULSE WATER WHEEL

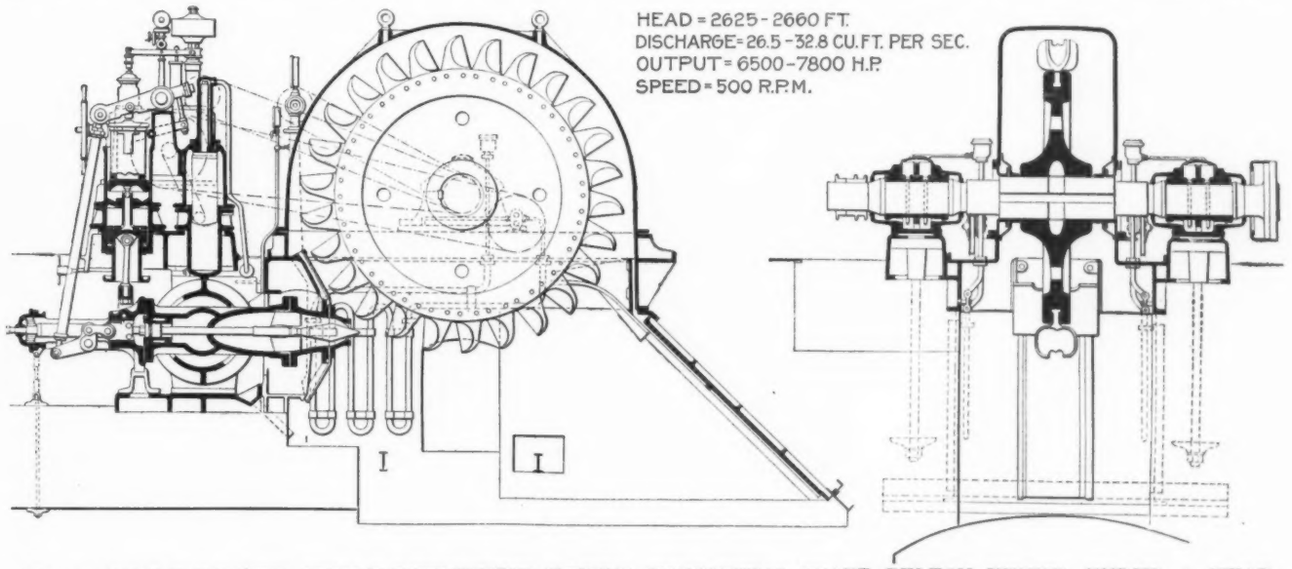


FIG. 8. SINGLE-RUNNER, SINGLE-DEFLECTING NOZZLE, HORIZONTAL-SHAFT PELTON WHEEL UNDER A HEAD OF 2,660 FT. ON THE FLAMISELL RIVER, SPAIN

AUTOMATIC SPEED-GOVERNING DEVICES

About 25 years ago the governing of the turbine, that is to say the maintaining of a constant speed for every load, was accomplished by a mechanical governor. The servomotor

of the governor was controlled by a pendulum affected by every small variation of speed. The water pressure available at the turbine itself was used for operating the servomotor where possible. The turbines installed for working under medium heads were first equipped with governors provided with automatic servomotors. The turbines of 3,200 hp. installed in 1908 in a Swiss plant and working under a head of 170 ft. were equipped with water-driven servomotors. To avoid undue wear of the governing valves only the purest water can be used for such a purpose. It was therefore necessary to provide reliable working filters or to use, as in some plants, water from other sources. Of course the standard practice has been to filter the water itself of the turbine. Notwithstanding these precautions, the wear on all the regulating valves in such hydraulic governors was great. The eddies in the servomotor, which could hardly be avoided even in the best designs, also caused wear on the cylinders, bushings and stuffing-boxes, so that it was decided to use another liquid for the purpose, the more so as all the low-head plants could not yield pressure water themselves. The necessary water under pressure ought to have been supplied by separately installed pumps. As soon as pumps were found to be necessary and unavoidable it was obvious that oil was preferable to water. The first low-head water-power developments in Switzerland had already been equipped with oil-pressure governors. The elements for these are of course the same as for the hydraulically operated governors.

It is important to reduce to a minimum the cost of this additional apparatus, especially for the smaller plants. The pressure of the oil is produced by pumps, generally driven from the main turbine shaft. Some of them are even connected with the pulley of the pendulum. The pumps of the larger governors are often driven by small water turbines. The servomotors are either symmetrical or differential, the former being chosen for the smaller and the latter for the larger governors. The smaller governors are generally equipped with pumps large enough to supply three or four times the necessary quantity of oil required for one complete movement of the servomotor.

The differential piston pump has been adopted for the larger governors. Two advantages are thus obtained. The pump needs only a capacity of one-third to one-fourth of the total volume of the oil required, the remainder being kept in a reservoir with an air vessel. The smaller end of the differential piston is constantly under pressure, preventing the turbine being started before the governing valves have acted.

The smaller oil-pressure governors have been recently improved by allowing a clearance on the governing piston, instead of an overlap. With the symmetrical servomotors the pressure of the pump is reduced almost to zero when the load is constant on the turbine, and the full working pressure is supplied by the pump only when the guide vanes of the turbine are to be moved. The advantage of this design is reduced heating of the oil during the operation.

The present tendency in oil-pressure governor design is to reduce the closing time of the governors; that is, the time necessary to shut down from full- to no-load. To insure small variations of velocity when taking off the load the flywheel momentum of the turbine is reduced in a rate proportional to the closing time. The controlling gears of the governor are made as sensitive as possible. The standard relay mechanism of the governor provides for a variation of three to four per cent. of speed between no-load and full-load, the speed at no-load being higher. In practically all the governors it is possible to change the range while the governor is running and it is even possible to reduce this variation to zero. This is the so-called isochronous regulation, which has already been tested when starting the large turbine plant at Niagara Falls. It is also possible to have a negative range of velocities, the speed increasing with the load. The modern oil-pressure governor is an extremely ingenious mechanism requiring the highest class of workmanship and is the essential factor that renders possible the installation of large power stations under the control of reduced staffs.

Massachusetts Board of Boiler Rules Hearing

The semiannual hearing of the Massachusetts Board of Boiler Rules was held Thursday, Nov. 4, at the State House, Boston. Several amendments were proposed, the one concerning which there was the most discussion being that on steel castings.

Superheater parts and water legs and door-frame rings must be of wrought or cast steel to conform to the present rules (amendments 24 and 25) and the tensile strength (amendment 42) is specified as 50,000 lb. minimum and 60,000 lb. maximum. In the two amendments offered to this amendment (No. 42) it was stated that foundries would not guarantee cast steel between these limits of tensile strength. As an example of how difficult it is to get cast steel to conform with the specifications, Mr. Brown, of the Power Specialty Co., stated that the Crane Co. offered to make such steel provided the customer would agree to take the whole of the steel casted which of course is not commercially feasible. It was recommended that the following be allowed as minimum tensile strength, no maximum being specified, and that the sulphur content be kept at 0.05 per cent. with 20 per cent above as a permissible variation: Hard, 80,000 lb.; medium, 70,000 lb.; soft, 60,000 lb. Mr. Jeter, chief engineer of the Hartford Steam Boiler Inspection and Insurance Co., approved the proposed amendment, saying that there should be no upper limit to the tensile strength, and that if the steel conformed to the chemical requirements the tensile strength would take care of itself. A representative of the Penn Steel Casting Co. recommended the adoption of the specifications of the American Society for Testing Materials for cast steel. These specifications are in use by the Federal Government.

Much discussion was had on the drilling of stay-bolts. The rules now specify that holes in stay-bolts, when such holes are less than ½ in. diameter, shall be drilled. Conforming to this rule, many present said, imposed an unnecessary hardship on boiler makers when drilling had to be done on stay-bolts from the inside of the boiler. There is little room to work, and drills break off, causing considerable delay. Further, it was said that the breaking of a stay-bolt on the inside was a rare occurrence. Also, when a stay-bolt is drilled from the inside to conform to the rules, it is not drilled all the way through. There is no possibility of properly cleaning the hole of the soot that it is sure to collect, and when soft coal of a high sulphur content is used there is a tendency to corrosion as soon as moisture gets at the soot in the bolt. An amendment was offered to allow the use of hollow rolled iron for stay-bolts instead of using solid ones and drilling them.

H. C. Earl, of the Walworth Manufacturing Co., proposed an amendment concerning fusible plugs. He argued that a ½-in. plug with 14 threads per inch did not permit of enough threads holding the plug when it was put in a boiler tube, and recommended that a ⅜-in. pipe-size plug containing a ⅜-in. hole for the fusible metal be allowed, instead of the ½-in. plug with ⅜-in. hole for the fusible metal, which is now allowed. The contention is that the outside diameter of a plug with ⅜-in. pipe threads will give plenty of metal around the ⅜-in. hole which contains the fusible tin. The ½-in. pipe-size plug has but 14 threads to the inch, and on a tube (the smaller the diameter the worse) there are not enough threads engaged to give good holding power and insure tightness of the plug. The ⅜-in. pipe-size plug would have 16 threads per inch, giving plenty of holding power and would contain a ⅜-in. hole that would conform to the present hole-size requirements.

Mr. Earl's amendment carried further provisions relative to plug dimensions. The rules specify that plugs must project through the sheet not less than 1 in., and on page 55 of the 1915 edition are given drawings and dimensions of plugs that do not insure that the projections will be had. Also, to conform to the rules relative to outside plugs, manufacturers must make a longer plug than is used to accomplish the same purpose which is specified by the United States supervising inspectors of steam vessels. This compels carrying two different plugs for the same purpose and charging more for the longer plug called for in Massachusetts.

Another proposed amendment by Harry E. Morton, of the Boiler Inspection Department of Massachusetts, provides that material covering all lap seams shall be removable and be removed by the owner at inspection times. This is to permit examining the seam at every inspection and likely detecting conditions that might otherwise go on until a dangerous state of the joint was reached. Objection was raised by Mr. Stewart, of the Dillon Boiler Works, that such legislation would be discriminatory and that the amendment should include all kinds of boiler joints. This was the general opinion, too, though Mr. Morton explained that his desire was to give attention first to those joints which needed it most.

Thomas E. Durban, representing the Boiler Code Committee of the American Society of Mechanical Engineers, requested that the board appoint a committee to confer with a subcommittee of the Code Committee to adjust the differences existing between the A. S. M. E. code and the Massachusetts rules.

Mr. Chapman, representing the New England Association of Heating Contractors, protested against the ambiguities of the rules relative to boiler fixtures and how and where they should be attached.